The ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

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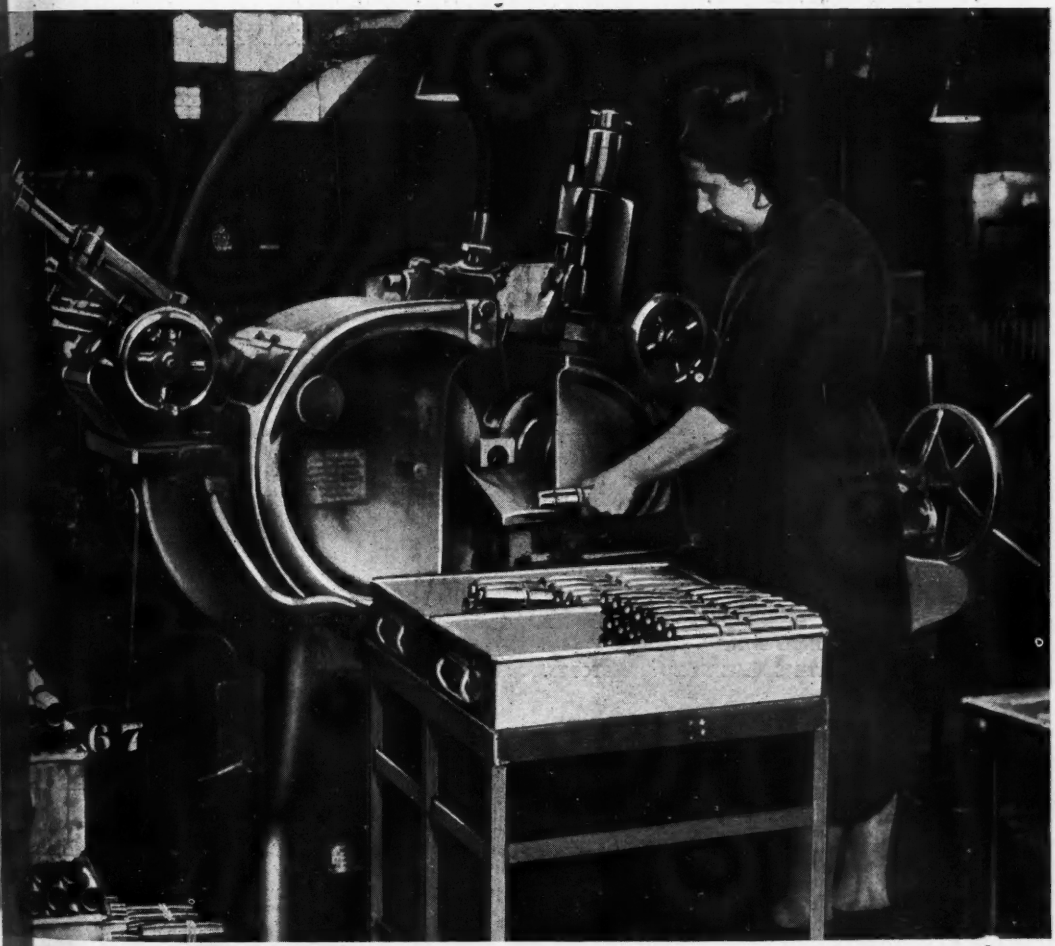

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RESEARCH AND THE WAR EFFORT

By VANNEVAR BUSH, F.A.I.E.E., Director of OSRD, President of Carnegie Institution of Washington (D.C.).

(From an Address delivered at the "Get-Together" dinner of the A.I.E.E. National Technical Meeting in New York on January 26th, 1943).

To a democracy, nothing is more important in the conduct of a war than the working out of smooth relationships between military and civilian organizations in all aspects of the war effort.

It is not sufficient to be possessed of good weapons; it is essential to be possessed of weapons that are better than those in the hands of the enemy and to use them with greater skill. The attainment of this objective involves the closest sort of collaboration between military men on one hand and civilian scientists and engineers on the other.

I believe that we have gotten on more rapidly and more effectively during this present war in the development and introduction of new weapons under the form of organization whereby civilian groups supplement the work of the Army and Navy than we would have had the entire affair been closely under military control.

New and valuable ideas are much more likely to come to fruition if they can develop their formative stages among groups of independent scientists and engineers before being subjected to the rigors of military association. We are engaged with skilful and resourceful enemies, and we should not at any point underestimate them. Germany in particular has too long a history of scientific and technical accomplishment for us to be tempted to underrate its possibilities in applying its skill to the conduct of war; and Germany has been fully engaged in the development of war techniques for a much longer time than have the democracies. Nevertheless, I believe that the present rigid military regime in Germany is at a disadvantage, when it comes to the development of really new ideas, as compared with the United States, with its ingenuity and resources; and I believe that is especially true in view of the fact that the organization under which we operate gives full rein to the independent efforts of some of the finest scientists and engineers that the country has produced, under conditions in which they can work substantially in their own way and in accordance with their chosen methods, but toward the common end.

But a democracy in wartime has certain handicaps also. Unfortunately, from one point of view, our own country has a striking difficulty in adapting itself to modern war. We think in terms of mass production. This constitutes one of our greatest factors of strength. It has, however, the distinct liability that we are likely to think in terms of freezing of designs and production of great masses of standardized equipment, and we think less readily in terms of a rapidly changing technical situation. In modern war it is a serious thing to be inflexible in this regard.

Under ordinary peacetime circumstances the progress from a brand new idea to its use in quantity by the public occupies at least five years. There has to be research, development, and engineering design. There has to be design for production and user experience obtained under carefully controlled conditions. Out of this can come a well-engineered device adapted for production in quantity to meet a mass need in an economic manner. Under ordinary peacetime conditions a company that is introducing a new product will short-circuit this proper and deliberate method at its peril, for large indeed are the penalties of plunging into quantity production before all the loose ends are tucked in. Yet in time of war we are faced with the dilemma of shortening this process or else being dangerously behind the times. Under the stress of war it is possible to compress the time scale somewhat. If it is compressed too much, the results may be very sad. On the other hand, delay in getting new devices into operation in time may have consequences that are disastrous. The attainment of a proper balance in this regard is one of the most difficult problems confronting the industry that produces the devices and the military groups that utilize them.

In the progress of a new weapon, from the first idea, to the final use, engineers and scientific men of professional grade enter at many points. Notably they appear as a part of the personnel of the armed services themselves, and they appear also in industry in the manufacture of equipment, and also in those services which are auxiliary to manufacturing effort, but none the less essential if the whole scheme is to function adequately. I feel that it will be worth while to trace the phase that is concerned primarily with the development of the new weapon from the standpoint of the governmental organization which has been charged with this responsibility in this present war.

In June, 1940, there was formed the National Defence Research Committee, charged by the President with the duty of research and development on new weapons and instruments of war. The initial organization was relatively small, but it has grown to a considerable scale. In June, 1941, a reorganization occurred and the Office of Scientific Research and Development was formed by executive order. The OSRD was given the broad task of co-ordinating the efforts of scientists and technical men in connection with many phases of the war effort, but it was also given the definite charge of pursuing aggressively the work that had already been started by the NDRC; and for this purpose NDRC was incorporated into its organization.

At the present time the OSRD is making expenditures at the rate of about $100,000,000 a year. In terms of the over-all war cost this is not a large amount of money. It is, however, a substantial sum when considered in terms of research and development. The OSRD operates entirely by contracts with existing academic institutions, industrial organizations, and government agencies. This method is designed to utilize to the utmost available facilities and personnel and to avoid, as far as it can be accomplished, the construction of great new laboratories. About 2,000 contracts for the carrying on of research have now been entered into, and of these about 1,400 are currently active. Approximately 200 industrial laboratories and 100 colleges and universities are at work on OSRD projects, and the number of men involved of professional grade is in the neighbourhood of 6,000. These men have assistants of various types.

There is the closest sort of inter-relationship with the Army and Navy at all levels. The NDRC is broken down into 18 major divisions concerned with various phases of war research, and in many cases these in turn are subdivided into sections. The research projects arise in these sections, usually by reason of round-the-table discussion of the current situation and the needs for improvement, and out of such discussions comes usually a definite request from either the Army or the Navy that the OSRD undertake a development along a certain line.

The NDRC examines the operations of divisions

through the medium of small subcommittees. It is the central reviewing agency which ties together the entire programme. Dr. James Conant, President of Harvard University, Boston, Mass., is the Chairman of NDRC, and its members are as follows :

Doctor Roger Adams, Head of the Department of Chemistry of the University of Illinois, Urbana ; Doctor Karl T. Compton, President of the Massachusetts Institute of Technology, Cambridge, Mass. ; Doctor Frank B. Jewett, President of the National Academy of Sciences ; Doctor Richard C. Tolman, Dean of the Graduate School of the California Institute of Technology, Pasadena.

These are the men who primarily represent American science and engineering. The Honourable Conway P. Coe, Commissioner of Patents, is also a member and brings to the committee his wide knowledge of inventions and their appropriate handling. The Army is represented by Major-General C. C. Williams. Captain Lybrand P. Smith represents the Navy Department.

The director of OSRD also has the benefit of an advisory council, which comprises : Harvey H. Bundy, Special Assistant to the Secretary of War ; Rear Admiral J. A. Furer, Co-ordinator of Research and Development of the Navy ; Doctor James Conant, representing the NDRC ; Doctor A. Newton Richards, Chairman of the Committee on Medical Research ; and Doctor J. Hunsaker, Chairman of the National Advisory Committee for Aeronautics.

It has also recently included in its discussions Doctor Harvey N. Davis, Director of the Office of Production Research and Development of the War Production Board. By special direction of the President, the director of OSRD has the benefit also of the advice of the President of the National Academy of Sciences, who joins in many council deliberations.

The business affairs are handled by the executive secretary of OSRD, Doctor Irvin Stewart.

The organization inludes also a liaison office, charged primarily with the duty of conducting appropriate technical interchange with the Allies of the United States. Under specific instructions from the President, there was instituted very early a close interchange with the British on technical matters, and this relationship has continued in a cordial and effective manner. I feel sure that this interchange has expedited the work of scientists and technical men in England in their magnificent efforts for the protection of the British Isles, and I am sure that it has benefited the United States in its war effort. The liaison office, under Doctor Caryl P. Haskins, maintains a London office through which there are contacts with the British Government at all times.

In order to appreciate the way in which the OSRD operates, it is necessary to realize that practically everything he does is highly secret. For reasons of security, appointment in personnel in any capacity throughout the organization is made only after careful investigation. A ruling principle, and one which is observed by the Army and Navy, is that secret matters are held carefully in compartments. This means that no member of the organization will learn of secret matters except to the extent that is necessary for his appropriate functioning in the particular position which he occupies in the organization.

While OSRD is organized nationally, drawing its membership from all over the country, its sections are made up of men especially adapted for the problems before them, and these men are given full knowledge of the technical and tactical phases of the particular weapons with which they deal. They are kept closely in touch with the progress being made in introducing weapons of this particular type in practice, and they form teams which are able to enlist the services of large numbers of men in many universities and industries for the accomplishment of their purposes. All this is done in such a manner as to keep the secret information as closely confined as is consistent with rapid progress.

As previously mentioned, not all of the research and development on weapons in this country is carried on by OSRD. Both armed services maintain large laboratories in peace and in war for the development of weapons, and they also further development by direct contract with industry.

There are several aspects of the development of new weapons which very definitely do not come under the control of OSRD, although the council of OSRD functions in an advisory capacity in order to provide a unitary approach to problems of common interest and to prevent overlap and duplication.

Notable in this connection is the National Advisory Committee for Aeronautics. Founded by Congress over 25 years ago, it has a long and notable record of accomplishment. Inasmuch as the problems of flight are being attacked adequately by the NACA, they are not again attacked by the NDRC, although the latter often carries on work on military devices which become incorporated in aeroplanes. It is a fortunate thing for the United States that it has had for many years an active research organization in the field of aeronautics. It has worked in close collaboration with the Army and Navy and with industry. A short time ago there was a great deal of discussion as to whether American aeroplanes were comparable with those of the enemy. Since the records have been coming in from England, Africa, and the South Pacific, this discussion seems to have become resolved.

The ideas that finally become incorporated in new military devices originate in a great variety of ways. Many of them come directly from officers of the armed services, as our combat indicates to officers at the front the needs and opportunities. Some ideas come from the scientific and military groups assembled as sections of NDRC as a result of their conferences and discussions. A large number are submitted by the general public. These require a great deal of review which is provided by the National Inventors Council.

The NIC is the official reviewing agency, and its function is fully performed when it has brought a valuable suggestion to the appropriate attention in the armed services. When the armed services find that an idea warrants development, they may turn to the OSRD in order to have such development performed.

There is another phase of the work of scientists and engineers on developmental matters which also needs to be mentioned. There are broad problems of substitute and strategic materials, and many technical questions involved in the reorientation of industry to the war effort. The development of substitute materials and substitute processes is of enormous significance as the war proceeds. The task of conducting research and development along these lines is not within the scope of OSRD, the activities of which are directed to the development of new weapons and their methods of utilization. On the other hand, the War Production Board has long been deeply concerned with these very matters. In some of its approaches to the problems of materials it has been strongly supported by the National Research Council with scientific and technical advice. Recently, a new office within the WPB has been formed, called the Office of Production Research and Development, and Doctor Harvey N. Davis, President of Stevens Institute of Technology, Hoboken, N.J., is director of this new office.

Many auxiliary problems arise in connection with the scientific and technical effort, and prominent among these is the problem of trained personnel. Fortunately,

we have had for a long time now the Roster of Scientific and Specialized Personnel, conducted under the chairmanship of Doctor Leonard Carmichael, President of Tufts College, and his office is now attached to the War Manpower Commission, where it is rendering excellent service.

The rapid survey of the organizational set-up would not be complete, however, without final mention of the National Academy of Sciences and the National Research Council. The National Academy of Sciences was formed at the time of the Civil War and operates under Congressional charter. It is charged with the broad duty of advising agencies of government with regard to their scientific and technical problems. The OSRD leans on the Academy and Council for scientific and technical advice on many matters.

Results of Research and Developments.

But what of results? In wartime the work of the laboratory is meaningless unless it finds its way into the field of action. The most tangible expression of the success of scientific effort in the present war is to be found in the attitudes of the armed services. Both the Secretary of War and the Secretary of Navy have indicated to me their satisfaction over what joint efforts have accomplished in terms of operations. At the close of the last fiscal year, the War Department pointed out that it had placed orders amounting to approximately $560,000,000 for items developed by one section alone of the NDRC. As a result of a new process for making an important military material developed by another section, the Army placed orders for plant and product amounting to about $270,000,000. In this instance there was an anticipated initial saving of $100,000,000 in plant-construction costs and additional savings of many thousands of dollars a day in operating costs, as compared with previous methods.

These figures are now some months old, and a great deal has happened since then. When the story can be told, it will be dramatic, and it will reflect the vigorous efforts of a great group of men, employing the best of team-work in the common cause. Until then we cannot talk of results. The evidence accumulates rapidly, however, that the devices being developed by American scientists and engineers will play an important part in bringing the war to successful conclusion in a shorter time than might otherwise have been the case.

THE INFLUENCE OF DEFORMATION ON THE INTERNAL LOADS OF BEAMS WITH A STRAIGHT AXIS

By F. REINITZHUBER. (From *Luftfahrtforschung*, Vol. 19, No. 9, October, 1942), pp. 320-325).

1. Introduction.

WHEN a beam with an originally straight axis is subjected to bending the axis deforms, and the curve taken by it is called the elastic curve. If, on the deformed beam we apply to two closely adjoining cross-sections the stresses in accordance with the ordinary theory of bending, $\left(\sigma = \frac{M}{I}y,\ \tau = \frac{Q\,M_{stat}}{Id}\right)$ these will, in general, not be in equilibrium, as the deformation of the beam axis has been neglected.

In the following article the influence of this neglect will be investigated. Beams with thin webs and two flanges will be considered only, such as are common in wing spars.

2. Calculation of internal Loads of Beams according to Common Bending Theory. (Deformation Neglected).

The beam is shown in Fig. 1. The cross-sectional

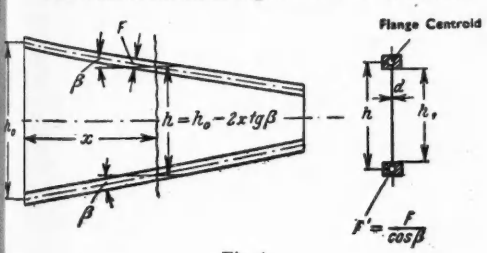

Fig. 1

area of each flange is F, the web thickness d. Due to inclination F of the flanges the distance of the flange centroid from the beam axis is $h = h_o - 2x \cdot \tan\beta$. In a cross-section of the beam the bending moment M and the shear force Q are acting (Fig. 2a). The moment of inertia of a beam cross-section is :

$$I = F' \frac{h^2}{2} + 2\,I_G + \frac{d\,h_1^3}{12},$$

where $F' = \frac{F}{\cos\beta}$ is the obliquely sectioned flange area,

and I_G the moment of inertia of a flange (both flanges assumed of equal area) about an axis through the centroid of flange, perpendicular to the plane of web.

Thus for the usual dimensions of such beams, the approximation :

$$I \simeq F' \frac{h^2}{2}$$

holds good.

Herewith, the direct stress σ_G at the flange centroid is :

$$\sigma_G = \frac{M}{I}\frac{h}{2} = \frac{M}{hF'}.$$

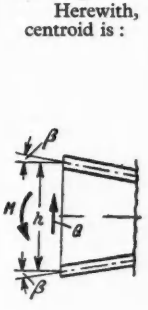

Fig. 2a

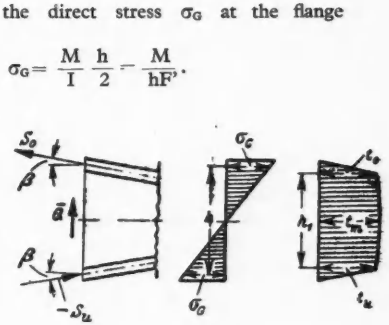

Fig. 2b

The axial forces in the flanges (Fig. 2b) are :

$$S_o = -S_u = S = \frac{\sigma_G F'}{\cos\beta} = \frac{M}{h\cos\beta} \quad \ldots \quad (1)$$

Due to inclination of the flanges the shear force Q changes to :

$$\overline{Q} = Q - 2\,S\sin\beta$$

and with due regard to eq. (1)

$$\overline{Q} = Q - \frac{2M}{h}\tan\beta \quad \ldots \quad (2a)$$

If the distribution of the shear force is according to the ordinary bending theory, the shear flux (Fig. 2b) becomes :

$$t_o = t_u = \frac{\overline{Q}\,F'}{I} \cdot \frac{h}{2},$$

$$t_m = \frac{\overline{Q}}{I}\,\frac{F'h}{2} + \frac{d\,h_i^2}{8},$$

or with d tending towards zero and $I = F'\,\frac{h^2}{2}$,

$$t_o = t_u = t_m = t = \frac{\overline{Q}\,F'}{I}\,\frac{h}{2} = \frac{\overline{Q}}{h} \quad .. \quad .. \quad (2b)$$

3. Calculation of Internal Loads of Beams allowing for Axis Deformation.

A point P of the originally straight beam axis has the co-ordinates x and y=o. Due to the bending moment and shear force the straight axis takes the shape of the elastic curve. Hereby, point P shifts to position P', u distant in the x direction, v in the y direction (Fig. 3). Assuming that the cross-section remains plane and perpendicular to the beam axis it must turn about angle φ, which is the inclination of the elastic curve.

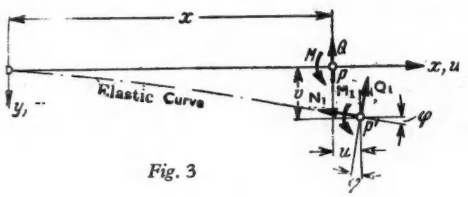

Fig. 3

The internal forces related to point P' are :

$m = M - u\,Q$ (Bending moment) .. (3a)
$q = Q \cos \varphi$ (Shear force) .. .. (3b)
$n = Q \sin \varphi$ (Axial force) .. .. (3c),

assuming that no change in magnitude and direction of external forces takes place due to deformation.

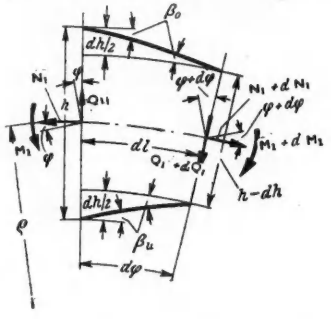

Fig. 4

A piece of the length dl cut out from the deformed beam is shown in Fig. 4. The height h of the deformed beam is assumed to be equal to the undeformed height.

Then the inclination β of the flanges to the beam axis changes ; for the upper flange which elongates :

$$\tan \beta_o = \frac{\frac{dh}{2}}{dl + \frac{h}{2}\,d\varphi} = \frac{\frac{1}{2}\,\frac{dh}{dl} \cdot \frac{dl}{d\varphi}}{\frac{dl}{d\varphi} + \frac{h}{2}} \quad .. \quad (4a)$$

for the lower flange, which shortens :

$$\tan \beta_u = \frac{\frac{dh}{2}}{dl - \frac{h}{2}\,d\varphi} = \frac{\frac{1}{2}\,\frac{dh}{dl} \cdot \frac{dl}{d\varphi}}{\frac{dl}{d\varphi} - \frac{h}{2}} \quad .. \quad (4b)$$

The forces m, n, q are changed by the differentials :

$$\left.\begin{aligned} d\,m &= -q\,dl = -Q\cos\varphi \,.\, dl \\ d\,q &= -n\,d\varphi = -Q\sin\varphi \,.\, d\varphi \\ d\,n &= +q\,d\varphi = +Q\cos\varphi \,.\, d\varphi \end{aligned}\right\} \quad (5a\text{-}c)$$

if one assumes that within dl there are no external forces.

According to the previous chapter and eqs. (3a-c) the force in the upper flange is :

$$S_o = \left(\frac{m}{h} + \frac{n}{2}\right)\frac{1}{\cos\beta_o} = \left[\frac{M - u\,Q}{h} + \frac{Q\sin\varphi}{2}\right]\frac{1}{\cos\beta_o} \quad (6a)$$

for the lower flange :

$$S_u = -\left(\frac{m}{h} - \frac{n}{2}\right)\frac{1}{\cos\beta_u} = -\left[\frac{M - u\,Q}{h} - \frac{Q\sin\varphi}{2}\right]\frac{1}{\cos\beta_u} \quad (6b)$$

and the shear force :

$$\overline{q} = q - (S_o \sin\beta_o - S_u \sin\beta_u)$$

or :

$$\overline{q} = Q\cos\varphi - \left[\frac{M - u\,Q}{h}\left(\tan\beta_o + \tan\beta_u\right) - \frac{Q\sin\varphi}{2}\left(\tan\beta_u - \tan\beta_o\right)\right] \quad (6c)$$

The differential increase of these forces in the upper flange :

$$d\,S_o = \left[\left(\frac{m + dm}{h - dh} - \frac{m}{h}\right) + \frac{dn}{2}\right]\frac{1}{\cos\beta_o}$$

or :

$$d\,S_o = \left[\frac{M - u\,Q}{h^2}\,dh - Q\left(\frac{dl}{h} - \frac{d\varphi}{2}\right)\cos\varphi\right]\frac{1}{\cos\beta_o} \quad .. \quad (7a)$$

in the lower flange :

$$d\,S_u = -\left[\left(\frac{m + dm}{h - dh} - \frac{m}{h}\right) - \frac{dn}{2}\right]\frac{1}{\cos\beta_u}$$

or :

$$d\,S_u = -\left[\frac{M - u\,Q}{h^2}\,dh - Q\left(\frac{dl}{h} + \frac{d\varphi}{2}\right)\cos\varphi\right]\frac{1}{\cos\beta_u}, \quad (7b)$$

with the shear force :

$$d\,\overline{q} = d\,q - (d\,S_o \sin\beta_o - d\,S_u \sin\beta_u)$$

or :

$$d\overline{q} = -Q\sin\varphi\,d\varphi - \left\{\left[\frac{M - u\,Q}{h^2}\,dh - Q\,\frac{dl}{h}\cos\varphi\right] \times \left(\tan\beta_o + \tan\beta_u\right) + Q\,\frac{d\varphi}{2}\left(\tan\beta_o - \tan\beta_u\right)\cos\varphi\right\}, \quad (7c)$$

whereby :

$$\frac{1}{h - dh} = \frac{1}{h}\left(1 + \frac{dh}{h}\right).$$

In separating the flanges from the web by cutting along the rivet line of the web to flange joint (Fig. 5), each of the three parts (both flanges and web) must be in equilibrium. The forces at the cuts are split into components t and p, parallel and perpendicular to the cutting plane respectively. The equilibrium for the upper flange together with eqs. (6a-b) or (7a-b) gives :

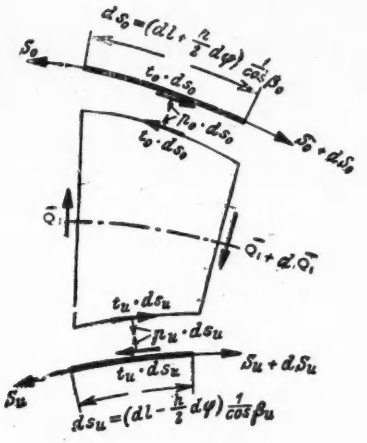

Fig. 5

$$t_o = \frac{d\,S_o \cos\beta_o}{dl + \frac{h}{2}d\varphi} = \frac{-\frac{dh}{dl}\frac{dl}{d\varphi}\frac{M-uQ}{h^2} + \frac{Q}{h}\left(\frac{dl}{d\varphi} - \frac{h}{2}\right)\cos\varphi}{\frac{dl}{d\varphi} + \frac{h}{2}} \quad (8a)$$

and:

$$p_o = -\frac{S_o \cos\beta_o}{\frac{dl}{d\varphi} + \frac{h}{2}} = \frac{\frac{M-uQ}{h} + \frac{Q\sin\varphi}{2}}{\frac{dl}{d\varphi} + \frac{h}{2}} \quad .. \quad (9a)$$

and that of the lower flanges yields:

$$t_u = \frac{d\,S_u \cdot \cos\beta_u}{dl - \frac{h}{2}d\varphi} = \frac{\frac{dh}{dl}\cdot\frac{dl}{d\varphi}\cdot\frac{M-uQ}{h^2} + \frac{Q}{h}\cdot\left(\frac{dl}{d\varphi} + \frac{h}{2}\right)\cos\varphi}{\frac{dl}{d\varphi} - \frac{h}{2}} \quad (8b)$$

and:

$$p_u = -\frac{S_u \cos\beta_u}{\frac{dl}{d\varphi} - \frac{h}{2}} = -\frac{\frac{M-uQ}{h} - Q\frac{\sin\varphi}{2}}{\frac{dl}{d\varphi} - \frac{h}{2}} \quad (9b)$$

The values $\frac{dl}{d\varphi}$ and $\frac{dh}{dl}$ are unknown in the above equations and have to be determined. The elongation $\Delta dx = d\mathit{l} - dx$ is due to the axial force n, when the length dx of the straight beam increases to $d\mathit{l}$ (Fig. 6).

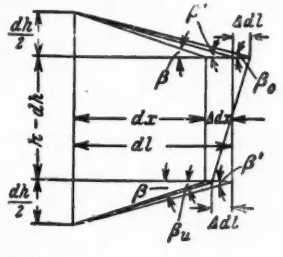

Fig. 6

By Hooke's law:

$$\Delta dx = \frac{n}{2EF\cos\beta}dx, \quad .. \quad (10a)$$

E is Young's modulus of the flange material, then by using eq. (3c):

$$dl = dx + \Delta dx = dx\left(1 + \frac{Q\sin\varphi}{2\,EF\cos\beta}\right) \quad .. \quad (10b)$$

By geometry, $$\frac{dh}{2\,dx} = \tan\beta \quad .. \quad .. \quad .. \quad (11)$$

and the eq. (10b) we obtain the required value:

$$\frac{dh}{dl} = \frac{2\tan\beta}{1 + \frac{Q\sin\varphi}{2\,EF\cos\beta}} \simeq 2\left(1 - \frac{Q\sin\varphi}{2\,EF\cos\beta}\tan\beta\right) \quad .. \quad (12)$$

Similarly, using Hooke's law and Fig. 6:

$$\Delta dl = \frac{m}{h\,EF\cos\beta'}dl$$

or, substituting eq. (3a) for m,

$$\Delta dl = \frac{M-uQ}{h\,EF\cos\beta'}dl \quad .. \quad .. \quad (13)$$

Angle β' is given by $\tan\beta' = \frac{dh}{2\,dl}$; herefrom and eq. (12):

$$\cos\beta' \simeq \left(1 + \frac{Q\sin\varphi}{2\,EF\cos\beta}\sin^2\beta\right)\cos\beta \quad .. \quad (14)$$

By geometry: $$\Delta dl = \frac{h}{2}d\varphi, \quad .. \quad .. \quad (15)$$

and eqs. (13) and (14) the second of the required values is obtained:

$$\frac{dl}{d\varphi} = \frac{h^2\,EF}{2(M-uQ)}\left(1 + \frac{Q\sin\varphi}{2\,EF\cos\beta}\sin^2\beta\right)\cos\beta \quad .. \quad (16)$$

Equations (6a-c), (8a, b), (9a, b) and (12) and (16) determine the stresses in the beam. However, a numerical evaluation is possible only if the inclination φ the elastic curve is known. This may be found by the usual method. The value u can, in general, be put equal to zero.

A comparison is afforded between the above theory and the simple bending theory which neglects the influence of deformation by putting:

$$u = 0 \text{ and } \varphi = 0 \quad .. \quad .. \quad (17)$$

The curvature $\frac{1}{\rho}$ of the beam axis must be zero as well. Now, $dl = \rho\,d\varphi$ (see Fig. 4), or $\rho = \frac{dl}{d\varphi}$.

Therefore: $$\frac{dl}{d\varphi} = \infty \quad .. \quad .. \quad (18)$$

If the calculation is followed to a conclusion it yields $p_o = p_u = 0$, that means that the beam dealt with in chapter 2 is a special case of the beam treated in this chapter.

4. The additional Radial Forces p_o and p_u.

It will be shown that these forces are not negligible. They are called here additional forces, because in the ordinary bending theory they do not appear altogether.

In substituting into eqs. (9a, b) eq. (16) for $\frac{dl}{d\varphi}$, one obtains

$$p_o = -\frac{S_o\cos\beta_o}{\frac{h}{2}\left(\frac{h\,EF}{M-uQ}\cos\beta' + 1\right)} = \frac{\frac{M-uQ}{h} + \frac{Q}{2}\sin\varphi}{\frac{h}{2}\left(\frac{h\,EF}{M-uQ}\cos\beta' + 1\right)} \quad (19a)$$

$$p_u = -\frac{S_u\cos\beta_u}{\frac{h}{2}\left(\frac{h\,EF}{M-uQ}\cos\beta' - 1\right)} = \frac{\frac{M-uQ}{h} - \frac{Q}{2}\sin\varphi}{\frac{h}{2}\left(\frac{h\,EF}{M-uQ}\cos\beta' - 1\right)} \quad (19b)$$

where $\cos\beta'$ is given by eq. (14). As u and φ are generally very small, we can write in eqs. (19a, b) $u=0$ and $\varphi=0$.

Further, considering that :

$$\frac{h\,EF}{M-u\,Q}\cos\beta' \gg 1,$$

the simple formula is obtained :

$$p=p_o=p_u\simeq-\frac{2\,M^2}{h^3\,EF\cos\beta} \quad .. \quad (20a)$$

or taking eq. (1) into account :

$$p\simeq-\frac{2\,S^2}{h\,EF}\cos\beta. \quad .. \quad .. \quad (20b)$$

For a parallel beam ($\beta=0$) :

$$p\simeq-\frac{2M^2}{h^3\,EF}=-\frac{2\,S^2}{h\,EF} \quad .. \quad .. \quad (21)$$

These additional radial forces are always compressive, as can be seen from Fig. 5 *. Whilst, in general, the calculation of the beam is always accurate enough when based on the ordinary bending theory, the additional radial forces cannot be neglected, as they are of the order of 10% or more of the shear force q. They can be determined for all practical purposes by eq. (20).

5. The Design of Beams with regard to the additional Radial Forces.

The existence of the additional radial forces p has a bearing on the design of the following beam parts :

(a) the web with its stiffeners
(b) the web to flange joint
(c) the flange.

*The negative sign is indicative that p is a compressive force. It should be noted that the German sign convention is contrary to British conventions.—Ed., E.D.

(a) As long as p is so small that the resulting stresses which it produces with the simultaneously acting shear flux are below the web buckling stress, these will be carried by the web. Then, it has to be checked, whether the reduced principal direct stress does not exceed the allowable stress.

If, however, the resulting stresses from p and the shear flux exceed the web buckling stress, the excess over the buckling stress has to be carried by the stiffeners. While determining, for instance, the compressive stresses in the stiffeners from the diagonal tension field, the forces p will increase the stiffener stresses.

(b) The dimensions of the rivet joint between flange and web are usually determined so that it carries the shear flux at the connecting line. Actually, the additional radial forces have to be carried too, and there is a slight increase in load per rivet (Fig. 7).

(c) If the flange centroid lies in the plane of the web, p has no effect on the flanges. An eccentric connection of the flanges to the web, however, sets up torsional moments in the upper and lower flange which are of opposite sign. (Fig. 8). They can be best accounted for by flexurally rigid stiffeners which join not only the web but the flanges too.

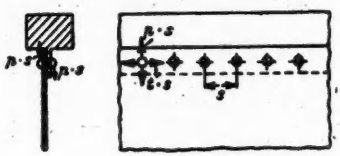

Fig. 7

Fig. 8

EXTENSOMETER MEASUREMENTS on the BACK AXLE OF A LORRY

By Dr. Ing. Habil. Ernst Lehr and Ing. Richard Schulz. (From *ATZ Automobiltechnische Zeitschrift*, Vol. 45, No. 17, 10th September, 1942, pp. 461-470).

(*Continued from page* 128)

In this way and with careful attention to detail, it was possible to reduce all sources of error to a minimum, so that in a large number of cases, where an independent check of the accuracy of the readings was possible, an error in the measurement of $\pm$ 1% to 2% was obtained. By special precautions a straight line calibration curve was obtained. The construction of the extensometer of 20 mm. measuring length can be seen from Figs. 9 and 10. The base b, made of brass, is mounted with its right end on the tapered pin f. It carries the horseshoe magnet a, which is built up of metal laminations of 0.05 mm. thickness. The magnet is carried in a frame l, which can be moved by means of a differential screw g, thus adjusting the air-gap. Any play in the thread of the differential screw is taken up by the spring k. After the adjustment of the air-gap, the frame l is clamped to the base by means of the screw h. The magnet is guided by a clamp m, fixed just behind the pole faces, which is held by means of fine wires, so that it gives way in a longitudinal direction, when the differential screw is being adjusted, but not, however, in a perpendicular direction. This guide prevents uncontrollable movements of the magnet, even with very great jerks.

The " keeper " is fixed to the brass lid which is mounted on a tapered pin at the other end of the measuring length. In order to ensure the relative position of the lid and the base, the guiding wires ensure parallel motion. The ends of the coils mounted on the horse shoe magnet are connected to a clamping board which is screwed to one side of the base. The flexible leads for the current supply consist of thin copper foil wound on cotton thread.

The construction of the comparison-magnet is shown in Fig. 11. The magnet a is screwed directly to the housing c of the instrument. The " keeper " is fixed to a cross-piece e, which is mounted on the lower end of the axle d, and is guided by slots in the housing c. The other end of the spindle d has a thread which screws into the differential nut f, the outside thread of which screws into the housing. The pitch of both threads differs only by 0.01 mm., so that one complete revolution of the differential nut causes a movement of the " keeper," relative to the magnet, of $\frac{1}{100}$ mm. only. Thus it is easily possible to adjust the air-gap to an accuracy of $\frac{1}{10{,}000}$ mm. The spring k removes the play in the screw threads by applying a force to the cross-piece e. The differential nut carries at one end a knob with a circular scale. A photograph of the assembled comparison magnet is shown in Fig. 12.

Fig. 13 shows a switchboard for four measuring circuits. The measuring circuits are separated from each other, so that one does not influence the other. The photograph of one element containing one measuring circuit is shown in Fig. 14. The adjustable resist-

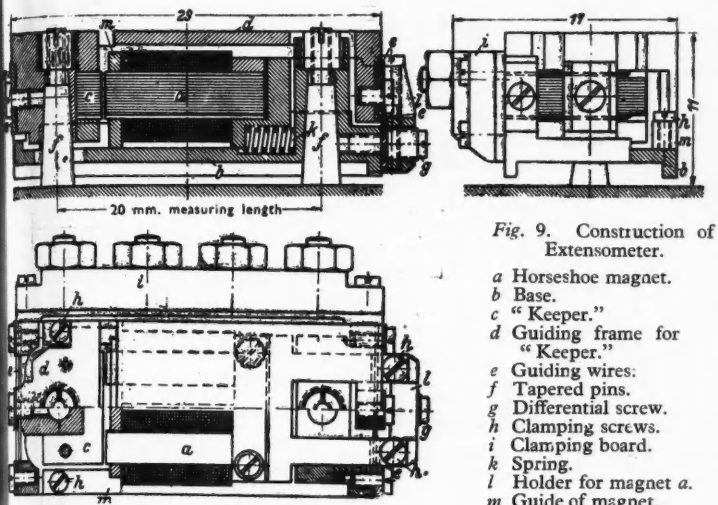

Fig. 9. Construction of Extensometer.

a Horseshoe magnet.
b Base.
c "Keeper."
d Guiding frame for "Keeper."
e Guiding wires.
f Tapered pins.
g Differential screw.
h Clamping screws.
i Clamping board.
k Spring.
l Holder for magnet *a*.
m Guide of magnet.

ance is completely separate to prevent overheating of the other instruments.

The tapered pins are soldered by means of a special electric soldering apparatus. Generally a zinc-cadmium solder is used. The soldering spots are heated to about 300° C, while the rest of the parts remain cold. The pins soldered like this can withstand a tensile load of 150 kg.; the pin can be bent at right angles without breaking the joint.

The extensometer is calibrated by means of a special instrument which can be adjusted to an accuracy of $\frac{1}{10{,}000}$ mm. The calibration is carried out in the laboratory for different air-gaps; in this way the determination of the calibration constant of the extensometer, when ready for use, is possible. For this purpose the "check-calibration curve" is determined, in which the current in the secondary circuit is plotted against the scale reading on the knob of the comparison-magnet (Fig. 15). The number of scale divisions on the knob necessary to change the secondary current from 3 mA. through the minimum on the calibration curve again to 3 mA. by turning the knob, is called "the opening" of the "check-calibration curve," which is a measure of the calibration constant. In the laboratory the "main calibration curve" is taken, in which the "opening of the check calibration curve" is plotted against the calibration constant.

Measurements and Results.

Static Loading of Back Axle.—The experiments were carried out by lifting the body of the loaded lorry in steps by means of a crane. A spring balance indicated the extent to which the load on the axle was decreased by the crane. The calculation of the bending

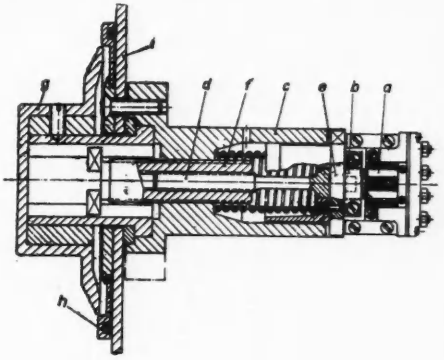

Fig. 11. Construction of Comparison Magnet.

a Horse shoe electro-magnet
b "Keeper"
c Housing
d Spindle
e Guide with cross-piece
f Differential nut
g Knob
h Dial
i Switch-board

moment exerted on the axle by the load Q through the two leaf springs can be understood from Fig. 16, the formula given below assumes that the axle is point-supported at the middle of the double wheels.

$M_b = a. \frac{Q}{2}$ cm. kg. The agreement between calculated and experimental values was accurate to 3%.

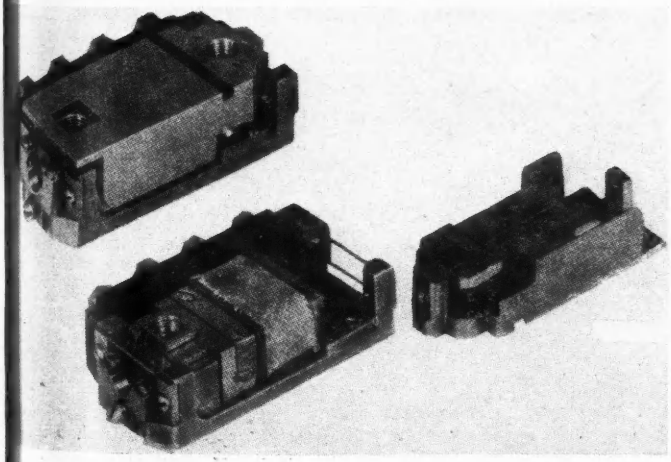

Fig. 10. Extensometer for 20 mm. measuring length.

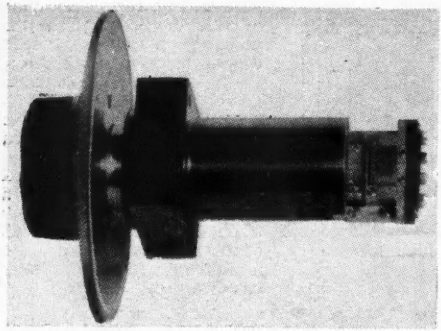

Fig. 12. Comparison magnet for 20 mm. measuring length.

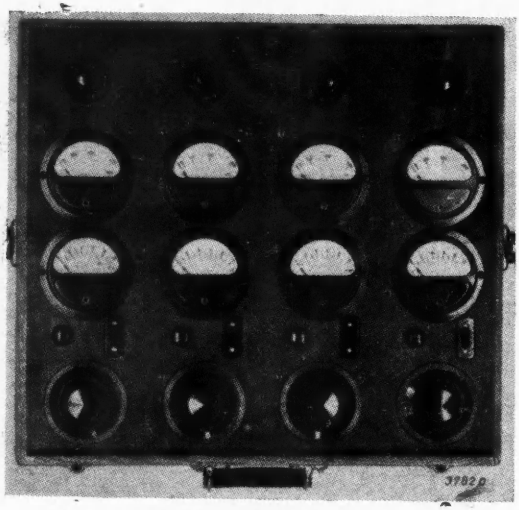

Fig. 13. Switch-board for four measuring circuits.

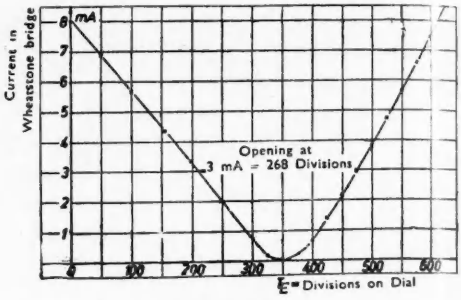

Fig. 15. Check calibration curve (specimen).

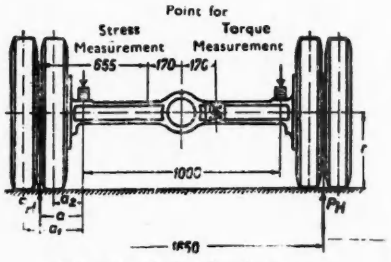

Fig. 16. Back axle of a lorry.

Method of Experiment.

To prevent shock due to uneveness of the road, the experiments of starting and braking were carried out as far as possible on the Reichsautobahn, between Nuernberg and Munich. The braking was carried out from speeds of 35 and 55 km/hr. Furthermore, experiments were carried out at starting from rest, and for each of the four gears. In order to investigate the effect of shock, experimental runs were carried out on badly paved roads at 55 km/hr. and on bumpy country roads at speeds up to 25 km/hr.

Evaluation of the Results.

This is done in the clearest way by constructing a diagram, as shown in Fig. 17. The extensions indicated

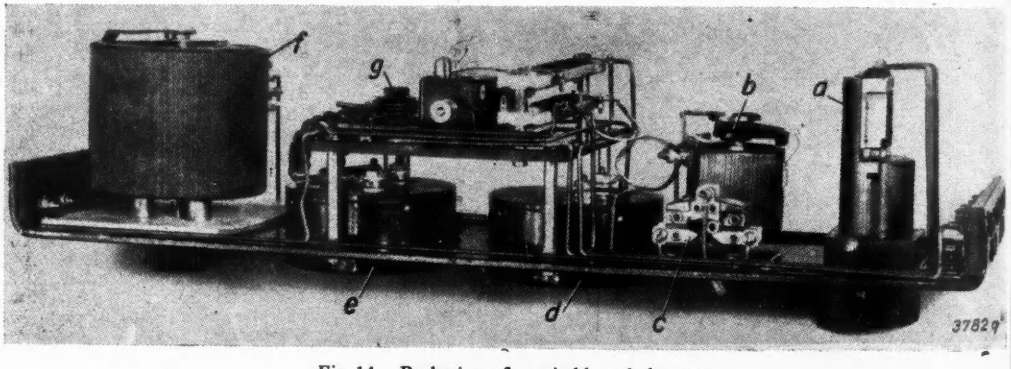

Fig. 14. Back view of a switchboard element.

a Comparison magnet; *b* Adjustable resistance for secondary circuit; *c* Safety switch for oscillograph coil; *d* Milliammeter for secondary circuit; *e* Milliammeter for primary current; *f* Adjustable resistance for primary current; *g* Copper oxide rectifier.

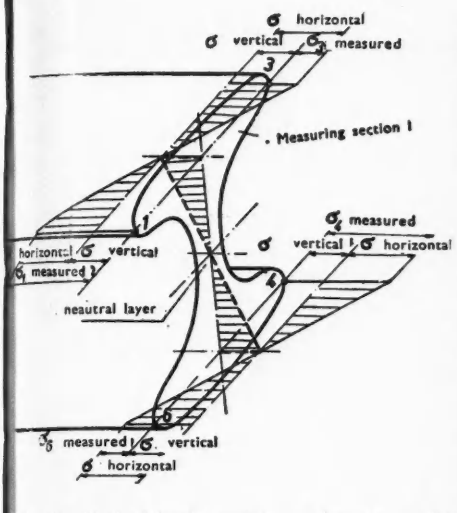

Fig. 17. Method for determining surface stresses from measured horizontal and vertical bending moments.

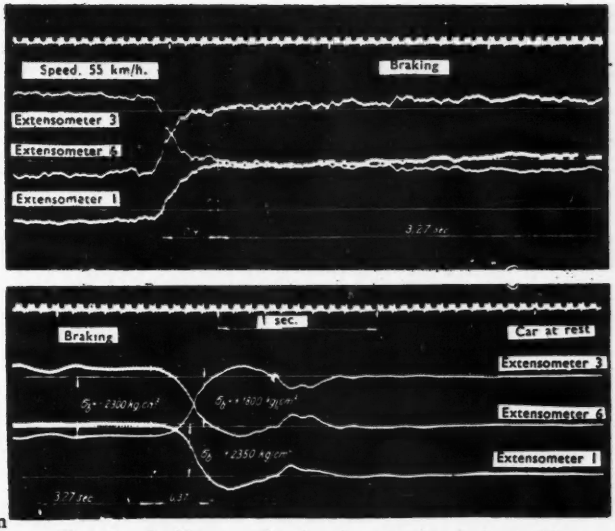

Fig. 18. Oscillograms showing the extension during braking.

by the extensometers 1, 3, 4 and 6, at the instant when the highest load is applied, are taken from the oscillogram. The surface stresses σ_1, σ_3, σ_4 and σ_6 are obtained by multiplying the corresponding strains by the modulus of elasticity E. Taking into consideration the sign, the stresses are plotted in the direction of the surface layer in which they act. Then the arithmetic means σ_1, and σ_3, and of σ_4 and σ_6, are formed. In the measurements performed these mean values were always approximately equal and of opposite sign. They are a measure of the bending moment acting in the vertical plane. A straight line is drawn through the end points of the lines representing the stresses mentioned above which intersects with the surface layer of the cross-section; the stresses obtained by this intersection must agree with the extensometer readings 2 and 5.

If the mean values are subtracted from the stresses σ_1, σ_3, σ_4 and σ_6 taking their sign into account, stresses are obtained, which, when multiplied by the corresponding section modulus, give the bending moment acting in the horizontal plan.

The measurements carried out in Sections II./III. give the torque acting in the axle. The evaluation of these results is very simple, as the oscillograph gives readings which are directly proportional to the torque.

Table I. contains the bending moments and torques obtained from the most important experiments. During the brake tests the maximum braking retardations were also determined. These are very important, as the bending moment acting in the horizontal plane is produced in the first place by the acceleration forces at starting, and the decelerating forces at braking.

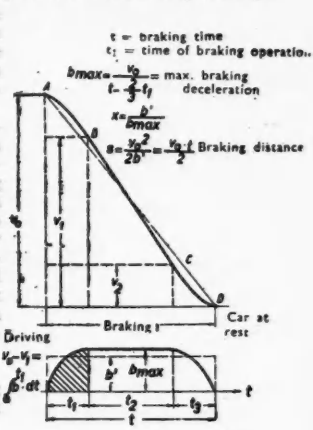

Fig. 19. Diagram of braking deceleration plotted against time.

TABLE 1. Values for the bending moments and torques, measured during changing gears, driving and braking on the German "Autobahn."

Test	Speed km/h	Bending Moment 1st Gear mkg W.	1st Gear mkg S.	2nd Gear mkg W.	2nd Gear mkg S.	3rd Gear mkg W.	3rd Gear mkg S.	4th Gear mkg W.	4th Gear mkg S.	Driving mkg W.	Driving mkg S.	Braking mkg W.	Braking mkg S.	Torque Gear 1 mkg	Gear 2 mkg	Gear 3 mkg	Gear 4 mkg	Driving mkg	Braking mkg	Braking Values Time t sec	Distance s m	Deceleration b' m/s²	Deceleration bMAX m/s²
Front and Back Brakes.																							
1	35	192	11,8	173	16	212	52	124	43	81	30	377	54,5	152	134	148	95	75	—53	2,4	11,6	4,04	4,44
2	40	286	42,0	260	95	270	44,5	194	120	143	76,5	325	32	185	182	157	112	69	—110	3,9	21,5	2,83	3,0
3	45	142	14,7	202	27,7	184	36	147	80	92	35,3	355	43,8	132	185	114	104	63	—47,5	3,1	19,4	4,04	4,3
4	55	136	25,6	162	47	120	58	130	39,5	63	24,4	330	33,6	144	162	131	140	—	+46	4,53	34,8	3,36	3,5
Back Brakes only																							
5	35	140	28,5	157	21,8	230	21	167	12,6	129	59	481	155	168	174	200	125	—	—	2,57	124	3,78	4,18
6*	35	223	63,0	153	18,5	218	38	155	26	103	20	514	128	190	144	134	100	53	+74	2,87	14	3,38	3,79
7	55	100	12,6	202	49	216	34,5	174	20	81	27,7	532	136	144	195	152	129	27	—31	4,3	33	3,55	3,77
8*	55	187	30,4	250	61,4	286	85,6	220	32,7	139	16,8	498	136	176	190	197	131	73,5	—28	4,04	30,8	3,8	4,06

* Clutch engaged during braking.

W = Bending moment in horizontal plan. S = Bending moment in vertical plane.

The Determination of the Brake-Deceleration.

A satisfactory approximation of the "brake deceleration" can be determined from the time-extension curve obtained from measuring Section I. It can be assumed that the recorded extensions are proportional to the corresponding brake decelerations. Fig. 20 shows that the time-extension curve during the brake deceleration is of parabolic shape. This fact is illus-

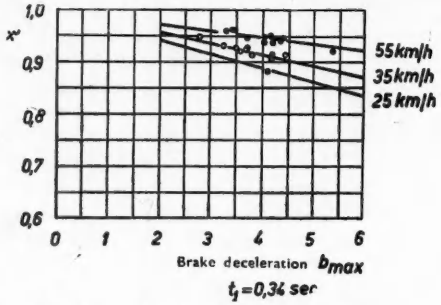

Fig. 20. Factor X plotted against maximum brake deceleration.

trated diagramatically in Fig. 19. Hence the following relations can be deduced :

If we assume that the time t_1 required for the beginning of the braking operation, equals the time t_3 required for the ending of the braking operation, then we can put

$$b_{max} = \frac{v_1 - v_2}{t_2}$$

where v_1 is the velocity at the end of the time t_1, and v_2 is the velocity at the end of the time t_2.

If finally v_0 is the velocity before braking, and v is the velocity at any instant, then

$$v_0 - v = \int b.dt. \quad \text{and also} \quad v_0 - v_1 = \int_0^{t_1} b.dt.$$

The integral is represented by the shaded area in Fig. 19, and equals two-thirds of the area of the circumscribed rectangle.

Hence $$v_0 - v_1 = \int_0^{t_1} b.dt = \frac{2}{3}\, t_1\, b_{max}$$

$$v_1 = v_0 - \frac{2}{3}\, t_3\, b_{max}$$

Similarly $$v_2 = \frac{2}{3}\, t_3\, b_{max}$$

Hence $$b_{max} = \frac{v_1 - v_2}{t_2} = \frac{v_0 - \frac{2}{3}\, t_1\, b_{max} - \frac{2}{3}\, t_3\, b_{max}}{t_2}$$

but as $t_3 = t_1$, and $t_2 = t - 2t_1$

$$b_{max} = \frac{v_0}{t - \frac{2}{3}\, t_1}$$

It is desirable to introduce the terms of the mean brake deceleration b_1. If t is the total braking time, then $b_1 = \frac{v_0}{t}$; in the diagram b_1 is the height of the rectangle with base t, the area of which equals the area below the curve b = f (t). Thus the following ratio X is obtained :

$$X = \frac{b_1}{b_{max}} = \frac{v_0}{t} \cdot \frac{(t - \frac{2}{3}\, t_1)}{v_0} = 1 - \frac{2}{3}\, \frac{t_1}{t}$$

Finally the braking distance

$$S = \frac{v_0^2}{2b_1} = \frac{v_0^2}{2x.\ b_{max}} = v_0\, \frac{t}{2}$$

In Fig. 20 the factor X is plotted against the maximum brake deceleration b_{max} for initial velocities v_0 = 55, 35 and 25 km/hr.

The values for b_{max} obtained on the "Autobahn" varied from about

4.2 to 5.4 m/s² at v_0 = 55 km/hr.
3.9 to 4.3 m/s² at v_0 = 45 km/hr.
4.0 to 4.4 m/s² at v_0 = 35 km/hr.

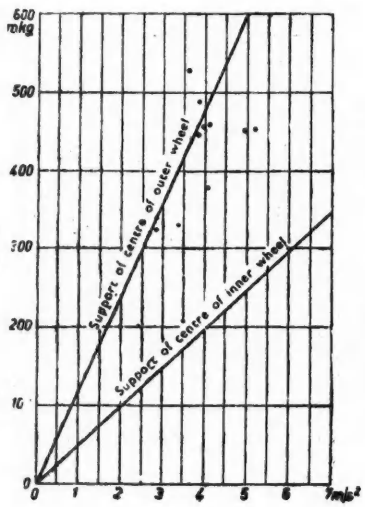

Fig. 21 (left).

Maximum bending moments acting in the horizontal plane, plotted against mean brake deceleration ; back brakes only.

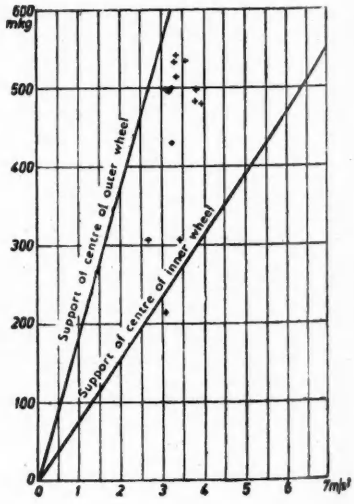

Fig. 22 (right).

Maximum bending moments acting in the horizontal plane, plotted against mean brake deceleration ; back and front brakes acting simultaneously.

In Figs. 21 and 22 the bending moment acting in the horizontal plane was plotted against the mean brake deceleration. In Fig. 21 only the back brakes were in operation, while Fig. 22 gives the results when both front and back brakes were operating simultaneously.

The measured values can be compared to calculated ones. In the case when the back wheel brakes only are operating, a force can be assumed to be acting in two equal halves on the axle at the points where the leaf springs are fixed; this force equals the product of the whole mass of the vehicle minus the mass of the complete back axle and the maximum brake deceleration b_{max}.

The axle, therefore, represents a beam supported at two points, loaded by two symmetrical forces. The question, however, arises as to where the two supports are acting. In Figs. 21 and 22 two extreme curves are shown drawn under the following assumptions:

(*a*) the support acts at the centre of the outer wheel;
(*b*) the support acts at the centre of the inner wheel.

It can be seen that the points obtained lie in between the two lines (*a*) and (*b*). The assumption (*a*), therefore, should correspond to a mean of the actual conditions; a considerable scatter should, however, be expected. The imaginary point of support evidently seems to shift from case to case. If the most unfavourable case that can occur is considered, it is advisable to base the calculations on assumption (*a*).

A CONTRIBUTION to the CHEMISTRY OF STEEL WORKS ANALYSIS

By A. STADELER. (From *Stahl und Eisen*, Vol. 62, No. 44, October, 1942, pp. 923-926.)

I.

MOST gas-washing bottles in common use have the disadvantage that they permit the flow of gas in one direction only. In order to wash the gas in series of flasks it is therefore necessary to disconnect the gas inlet and outlet. This constitutes an inconvenience especially if the gas is one which is harmful or badly smelling. There is also the danger that the wash bottles may accidentally be wrongly connected.

This defect can be easily avoided by the use of the gas-wash bottle headpiece introduced by W. Kwasnik[1]. The essential feature of this is its 4-way cock with bores all in the same plane. The two angle borings are independent of each other. Their opposite connections are in communication with the plunge tube of the flask exit tube, i.e., with the gas inlet and outlet. By simply turning the cock through 90° the direction of flow of the gas is reversed without disconnecting the apparatus. Turning the cock through 45° completely closes all exit tubes.

There is more and more tendency among modern analytical chemists to adopt physical methods which involve the direct measurement of the concentration in solution of the sample to be estimated. This usually necessitates adjusting the solution to a certain volume, 10 c.c. being usually sufficient. It is, of course, possible to use a cylinder or graduated flask for this purpose, but the large diameter necessary to allow of filling and emptying causes a large volume error. To obviate this a new mixing and measuring pipette has been designed by F. A. Uhl[2] (see Diagram 1). Space 1, up to the mark M, contains an exact volume, e.g., 10 c.c. Spaces 2 and 3 contain approximately the same. Filling should be done with the apparatus placed vertically in the stand. The stopper is removed and the liquid introduced with a pipette until it reaches the mark M. The stopper is now replaced and the apparatus inclined alternately from side to side so that the liquid flows into spaces 3, 2 and 1. This ensures a thorough mixing without the introduction of air. Emptying is done through spaces 3 and 4. Cleaning of this apparatus is simple and quick. As support a wooden funnel can be used.

For reasons of economy laboratory vessels such as crucibles, dishes and electrodes made of Pt or other precious metals have always been produced in a limited number of standard sizes and shapes. This policy is even more necessary now and the choice has been limited to a minimum of the most generally suitable shapes. The question of possible replacement of platinum by other metals has also been investigated Pure gold is too soft for hard wear and must therefore be alloyed with hardening agents such as copper or silver. The presence of these metals, however, considerably reduces the uses to which the crucibles can be put. The possibilities of a platinum-gold alloy containing 90% gold and 10% platinum which has for some time been known are now investigated.

Autoclaves constructed of Jena glass have been described by P. Schorning[3]. Glass does not share the two big disadvantages of metal for this purpose: (*a*) the tendency to corrosion, and (*b*) catalytic effects on the reactions. The new autoclaves are very easily cleaned, and their transparency permit close observation of their contents. The danger of super-heating, with its resultant subsidiary reactions, is also eliminated as the glass autoclaves can conveniently be suspended in a bath. They can be produced in all sizes.

W. A. Roth[4] describes a calorimetric bomb of Krupps V 2A steel which has now come into general use. A special feature is the automatic closure which protects the thread. The V 2A steel withstands even a high sulphur-containing material, especially if a little paraffin has been added. Material containing halogen should be burned only in a platinum-lined bomb. It is advisable to isolate the calorimeter and its water jacket by means of insulating leads of rubber or synthetic material. This bomb has the great advantage that standardisation is done by the same method as the estimations. Roth prefers cotton wool to metal wire for ignition. Its use saves weighing and estimation, and also decreases wear and tear on the bomb.

II. Crude Iron, Steel and Special Steel.

Of the many macro-analytical methods—that is, methods using a comparatively large sample—for the estimation of carbon in steel none is suitable for adaptation to micro-analysis. The chief problem in the estimation of CO_2 produced by ignition is the large quantity of air-oxygen by which it is always accompanied.

P. L. Gunther and V. Rebentisch[5] describe a precision micro method for the estimation of carbon in steel.

(1) Chem. Technik 15 (1942), pp. 122-123.
(2) Z. Analyt. Chem. 123 (1942), pp. 321-322.
(3) Chem. Technik 15 (1942), p. 46.
(4) Chem. Technik 15 (1942), pp. 63-64.
(5) Chem. Technik 15 (1942), pp. 17-18.

The sample is burned in the usual way in a stream of oxygen. Separation of the CO_2 from the excess oxygen is effected by cooling to a low temperature and pumping off the uncondensed material and measuring the quantity of CO_2 in a burette over mercury. Used as a micro method this procedure gives results whose accuracy equals those of the usual macro-analysis. Applied to macro-estimation it has an accuracy 50 times higher than that of other merhods. The whole process takes 30 minutes.

L. Silvermann and O. Gates[6] have evolved a method for the estimation of manganese, chromium and nickel in austenitic chromium-nickel steel. Silica and niobium are precipitated by perchloric acid after solution of the steel in aqua regia. Manganese is oxidised by persulphate to permanganic acid, which is titrated with arsenious sodium nitrite. Chromium is estimated with ferrous sulphate by the potentiometric method. Nickel is titrated with sodium cyanide and molybdenum colorimetrically.

G. H. Carledge and P. M. Nichols[7] have recently developed a spectro-photometric method of estimating cobalt as tri-oxalate-cobaltate. The method is much quicker than the usual gravimetric and electrometric ones. The compound is produced by oxidation of the cobalt, containing solution with lead dioxide in a slightly acidified solution of potassium oxalate. The tri-oxalate-cobaltate has a dark emerald green colour whose intensity can be used for estimation of the contained cobalt.

B. Mader[8] suggests the use of colorimetric analysis to estimate cobalt as cobalt thiocyanide. This can be carried out even by inexperienced workers, and takes only 10 minutes after solution of the steel. The cobalt thiocyanide is extracted by shaking the solution with amyl alcohol or acetone in which it dissolves a bright blue compound. This method is particularly suited to the estimation of the small amounts of cobalt, e.g., 0.005–2%, occurring mainly in nickel alloys, which are slow and difficult to estimate by gravimetric and electrometric methods. The only other coloured ions soluble in amyl alcohol are iron cyanide and, when it is present in great concentration, nickel. These elements, however, can be easily eliminated—iron by pyrophosphate, nickel by shaking with a mixture of amyl alcohol and ether. The other usual constituents of special steels, e.g., molybdenum, vanadium, aluminium and manganese, have no influence on the reaction. Tungsten should be kept in solution by the addition of phosphoric acid. Copper, by virtue of its colour, is a disturbing element. It should be present in quantity not exceeding 0.05 mgm.

For large amounts of cobalt such as occur in magnet-steel, e.g., 0.3–20%, it is advisable to work in an acetone-containing solution. Any iron present should be separated with zinc oxide.

A. Claassen and J. Visser[9] estimate zirconium as phosphate, taking into account the hafnium content. The zirconium is precipitated with ammonium phosphate in 10% sulphuric acid solution, which gives very accurate results. Hydrochloric acid may alternatively be used. Tartaric acid does not upset the reaction. Zirconium is sharply separated from aluminium, copper, cadmium, bismuth, nickel, cobalt, manganese, zinc, magnesium, the alkali earths and alkalis, tungsten, vanadium, molybdenum and uranium. Ferric iron is only completely removed if present in very small quantities. Larger amounts must be reduced to the ferrous condition. Titanium also is separated only in small traces. It has not been found possible to remove tin.

(6) Industr. Eng. Chem., Anal. ed., 12 (1940), pp. 518-519.
(7) Industr. Eng. Chem., Anal. ed., 13 (1941), pp. 20-21.
(8) Chemie 55 (1942), pp. 206-207.
(9) Rec. Trav. Chim., Pays-Bas, 61 (1942), pp. 103-119.
(10) Industr. Eng. Chem., Anal. ed., 13 (1941), pp. 71-72.

III. Ores, Slags and Fire-proof Material.

In the rapid estimation of ferro and ferric ions J. O. Percival[10] has introduced the use of precipitated copper powder as a reducing agent. This enables the estimation to be completed in 10 minutes.

W. Kriesel[11] describes a combined method for the volumetric estimation of iron, vanadium and chromium. It is particularly applicable to vanadium slags and ores and to ferro-vanadium. The three elements are reduced in sulphuric acid solution with hydrogen sulphide.

The iron and vanadium are titrated together with potassium permanganate solution. The vanadium in the titrated mixture is now again reduced with ferrous sulphate and retitrated with permanganate. The quantity used corresponds to the vanadium content and the difference on the two titrations represents the iron.

In the same solution the chromium is now oxidised by ammonium persulphate, and the chromium and vanadium together reduced with ferrous sulphate. A third titration with permanganate indicates the chromium.

For the separation of calcium and strontium nitrates H. H. Barber[12] recommends the use of mono-butyl-ether or ethylen-glyocol. He finds that both reagents give complete separation of these nitrates.

W. Hurka[13] has established a method for the acidimetric estimation of chloride. Its use is confined to chlorides in alkaline combination. Only a standardised solution of hydrochloric acid is necessary.

To the alkaline chloride solution silver sulphate is added until all the chloride has been precipitated. The alkali is now present as alkali-sulphate. Barium hydroxide is added to this in excess, forming a hydroxide of the alkali with an insoluble precipitate of barium sulphate, the excess ammonium carbonate and ammonia at the same time transforming the alkali-hydroxide into alkali-carbonate. The latter is titrated with standard hydrochloric acid, using methyl orange as indicator. From this titration the chlorine can be calculated.

M. Goehring[14] gives a new method for the estimation of sulphur in pyrites. The sulphur in copper pyrites is oxidised by a short period of heating at 600° to the sulphate. The mass assumes the form of a coarse powder but does not melt. The silica remains unchanged. The oxidised material is removed from the crucible and treated with hydrochloric acid, the insoluble material being filtered off. The sulphur is then estimated in the filtrate as barium sulphate. Besides its rapidity this method has the advantage of being unaffected by the presence of arsenic and antimony.

E. W. Koenig[15] suggests the estimation of aluminium oxide in certain silicates by precipitating the aluminium as the salt of an organic acid in the presence of silica. The precipitation should be preceded by alkaline fusion.

Fluorine is usually estimated as calcium fluoride, involving an extremely slow and difficult filtration after precipitation. H. Krause[16] has succeeded in obtaining the calcium fluoride in such a form that it can be easily filtered and washed over a vacuum pump. Instead of using a salt of calcium, the hydroxide, in the form of a watery suspension, is added to the neutral or nearly neutral fluoride solution.

(11) Metal u. Erz 39 (1942), pp. 143-144.
(12) Industr. Eng. Chem., Anal. ed., 13 (1941), pp. 572-573.
(13) Z. Anal. Chem. 123 (1942), pp. 165-168.
(14) Z. Anal. Chem. 123 (1942), pp. 399-403.
(15) Industr. Eng. Chem., Anal. ed., 11 (1939), pp. 532-535.
(16) Chemiker Ztg. 66 (1942), pp. 202-204.

THE INFLUENCE OF MOLYBDENUM & VANADIUM ADDITIONS ON LOW-TUNGSTEN HIGHSPEED STEELS CONTAINING 4% CHROMIUM

By HERBERT KESSNER. (From *Stahl und Eisen*, Vol. 62, No. 44, October, 1942, pp. 922-923.)

SOME time ago a high-speed steel was introduced in Germany with approximately the following composition: chromium 4%, carbon 1%, molybdenum 3%, vanadium 3.5%, tungsten 3%.

Extensive research has been done by A. P. Guljajew in Russia on steels containing 4% chromium, 3% molybdenum and 3% tungsten. He investigated the effect of the addition of 2–7% vanadium on the temper hardness produced in these steels by different tempering temperatures. Results showed that at a temperature of 1270° the steel which attained the highest temper hardness at 600° were those containing 2.3, 3.0 and 5.5% vanadium. 2.3% vanadium gave the best temper hardness at 650°; 4% vanadium the greatest hardness at 700°, i.e., approximately 32 HRc. The results of the turning experiments are given in Table I.

In steel with 4% chromium, 3% tungsten and 3% molybdenum the first 2.3% vanadium were most effective. Above this point the efficiency dropped again. The best vanadium content is, therefore, 2.5%.

By a standardised machineability test steel 1 with 17.6% tungsten showed a poorer cutting effect than steel 8 with its high vanadium content of 6.7%. Decrease of the vanadium to 1.7% in steel 3 caused a drop in cutting quality.

No decrease was produced by restricting the tungsten in steel 9 to 1.5%, but cutting down the molybdenum to 0.9% in steel 10 exerted some influence.

The optimum temperature for obtaining the highest cutting effect was found to be 1240°. A. P. Guljajew also examined the effect of additions of molybdenum in varying quantities up to 3% on the qualities of a tungsten-free high-speed steel containing 1.2% carbon, 4% chromium and 4% vanadium. His results showed that molybdenum increases the temper hardness and improves the cutting qualities of this steel. The most effective molybdenum content lies between 2 and 2.5%. The addition of more than 2.5% has no further influence.

Hardening at 1280° in oil, and tempering three times to 560°, produced a hardness of 64–65 HRc. The cutting qualities of these experimental steels on a steel with 220–230 H.B. were compared with those of a high-speed steel "R Fe," containing 0.71% carbon, 4.58% chromium, 19.38% tungsten and 1.2% vanadium. The turnings were 3×0.31 mm². The V_{11} figure—that is, the testing time—is approximately 34 m/min. This corresponds to that of a comparison steel containing 18% tungsten. The steel "E.J.277" (Z.G.S.) is, therefore, regarded as having the best cutting properties and is equivalent to a steel containing 18% tungsten. The analysis of this steel is as follows: 1.10–1.22% carbon, 3.8–4.6% chromium, 2.8–3.3% vanadium and 2.7–2.9% molybdenum.

The Russian experiments have been confirmed by German investigators. R. Scherer and H. Beutel found high qualities in a steel containing 1.12% carbon, 3.47% molybdenum and 2.63% vanadium.

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TABLE I.—Index of various high speed steels containing molybdenum and vanadium.

Steel No.	C %	Cr %	Mo %	V %	W %	Hardening Temp. ° C.	Tempering Temp. ° C.	Temper Figure	Machineability Test* V_{60} m./min.
*1	0.71	4.2	—	0.53	17.6	1300	560	3	42.5
2	1.05	4.8	0.9	3.6	3.1	1270	560	2	41.0
3	1.06	4.2	3.2	3.1	1.5	1270	560	2	44.0
4	0.76	4.1	3.1	—	3.2	1200	560	3	38.0
5	1.03	3.8	2.8	1.7	2.6	1190	560	3	42.5
6	0.94	4.1	3.0	2.3	3.1	1250	560	2	45.0
7	1.01	4.3	3.5	3.2	3.1	1270	560	2	44.5
8	1.17	3.9	2.9	3.0	3.2	1270	560	2	44.5
9	1.36	4.4	3.1	5.5	3.3	1290	560	2	44.0
10	1.45	4.0	3.3	6.7	3.3	1290	560	2	44.0

* V_{60} Rate of cutting a low-alloy nickel-chromium steel of standard hardness without cooling. Depth of cut 3 mm. and feed 0.31 mm.

THE FORMATION OF A ZINC-IRON ALLOY IN THE PROCESS OF HIGH TEMPERATURE GALVANISATION

By WILHELM RAEDEKER. (From *Stahl und Eisen*, Vol. 62, No. 18, 30th April, 1942, pp. 374-376.)

IN the investigation of the differences in properties and in the surface appearance of galvanised steel parts the following points are to be considered :

The precise alloy of steel to be galvanised, the duration of dipping, the temperature of the alloy or of the zinc bath, and the duration of cooling.

Experiments have so far failed to demonstrate the direct influence of any of these factors on the surface appearance, which appearance cannot yet be effectively controlled.

The properties of galvanised steels and the formation of a zinc-iron alloy during the galvanisation process have been investigated by altering two of the above factors, keeping the rest constant.

Research by H. Bablik and A. Merz has shown the importance of the silicon content of the steel. Keeping a maximum carbon content of 0.10%, a range of alloys was made which included silicon in percentages of 0.0, 0.7, 1.2, 2.5, and 3.5. The susceptibility of the alloy to attack by zinc is judged by the loss in weight of the foil (expressed in grams per square decimeter) after a dipping time of 1-64 minutes. Loss in weight in the first two stages of one-hour dippings is about 1.9, but this figure decreases with subsequent treatments. The explanation of the reaction is expressed in the following formula :

$$V = k \cdot t^n$$

where V = loss in weight, t = dipping duration, k and n are constants.

Where n is less than 0.5 diffusion has a strong influence on the reaction. When n is greater than 0.5 some part of the reaction product seems to find its way back into the zinc bath.

In the zinc plating practice it has been found that up to a maximum of 0.7% silicon the tendency of zinc and iron to form a compound increases with the silicon content of the steel. R. W. Sandelin has shown that the reaction is at its minimum when the silicon percentage is between 0.15 and 0.25.

W. G. Imhoff has contributed a paper on the formation of flowers in the galvanising of steel tubes. He found that raising the temperature of the bath increased the number while decreasing the size of the flowers. It seems that for the depositing of the very thin zinc layers which alone can produce these fine flowers, the formation of certain hard zinc crystals is essential. These crystals are formed most freely at high temperatures.

Prolonged dipping time also increases the number of crystals, and therefore the flower-formation. Conversely a short dipping time favours the formation of few and large flowers. Combination of a high temperature with a short dipping time gives a burnt appearance. The flexibility of the tubes is good.

As the bath temperature rises the deposited zinc increases. When a large number of tubes are placed in the galvanising bath and taken out separately, the tubes removed last are found to have a thick brittle coat of zinc with poor adhesion. To avoid this fault Imhoff advises the installation of a machine which coats the tubes separately.

It has been found possible to increase the polish and brilliance by the addition of the elements, notably aluminium and cadmium to the alloy. The influence of bath temperature and dipping duration are, however, dominant.

R. W. Sandelin has done a considerable amount of work on the relation of the composition of the steel alloys employed to the thickness, adhesiveness and appearance of the zinc deposited on it. Mettalographic methods further showed differences in the structures of the zinc coating.

Seventy-two steels were examined. They resulted from varying the quantities of one element in each of seven different meltings. The original melt included :

0.05 to 0.32% carbon
0.02 „ 0.29% silicon
0.27 „ 0.90% manganese
0.01 „ 0.07% sulphur
0.17 „ 0.39% copper
0.01 „ 0.10% phosphorus.

The contents of these elements were altered over the following range :

0.02 to 0.63% silicon
0.27 „ 0.90% manganese
0.01 „ 0.10% phosphorus
0.17 „ 0.68% copper
0.00 „ 0.17% titanium
Traces „ 0.30% aluminium.

It was found that the amount of copper, manganese, aluminium and titanium in the alloy had no influence on the quality or appearance of the zinc coating.

The best surface appearance was obtained from the use of un-alloyed steel.

Phosphorus and silicon caused, in a short time, greying of the coat. With prolongation of dipping the coat showed roughening. Phosphorus exercises a deleterious influence on the adhesiveness of the zinc, while silicon has the opposite effect unless the steel is rich in carbon, when the strength of attachment decreases with the carbon content.

The effect of phosphorus on the weight of the coat is dependent on the amount of silicon in the steel. If the silica constitutes less than 0.05% the thickness of the coat increases with a rising phosphorus content. With steels with a high percentage of silica, increase of phosphorus causes a decrease in the weight of zinc deposited.

The influence of the carbon content could not be observed.

With a mounting silicon content the weight of the coat increased up to a figure of 0.15%, decreased to a minimum between 0.20 and 0.30%, and then steadily mounted. The influence of the remaining elements was negligible.

General Remarks.

Dipping at high temperature produces a rough grain resulting in a thick grey covering instead of a fine clean coating of zinc. The fine and medium grain occurring in warm rolled steels has no influence on the weight or appearance of the coating. Cooling in water has no observed effect. Cementation increases the weight of the coat on silicated samples without altering their appearance. In unsilicated samples the surface was also unaltered ; the weight of the coating was dependent on the carbonisation. The operation confirmed the belief that the combined influence of carbon and silicon encouraged the reaction between iron and manganese. Carbonisation followed by hardening resulted in a bright smooth finish.

Summary.

Of the common constituents of soft steel alloys only sulphur and phosphorus appreciably influence the quality of the zinc coat. A silicon content of approximately 0.2% achieves the maximum adhesiveness and the best appearance. Dipping time should be as short, and bath temperature as low, as possible.

PROBLEMS IN AIRCRAFT STRUCTURAL RESEARCH

By F. R. SHANLEY, Chief Structural Engineer, Lockheed Aircraft Corporation, Burbank, Calif. (From *Mechanical Engineering*, Vol. 65, No. 3, March, 1943, pp. 169-178.)

IN so far as it is possible in wartime restrictions, the purpose of this paper is to present a comprehensive picture of the more important problems in aircraft structural design and research, in the interest of advancing the knowledge of those engaged in industries, which formerly had but slight connection with aeronautical engineering, but to-day are deeply involved in various phases of aircraft work. A similar objective was stated by Dr. A. G. Pugsley (1)* as follows :—

". . . By a discussion of the general engineering principles behind the structural work of aeronautics it is hoped at once to help newcomers to appreciate the present complicated position and the breadth of current problems, and to stimulate more experienced workers to the further clarification of the principles and problems involved."

Objects of Aircraft Structural Research.

First, let us ask ourselves why we spend time and money on structural research. The main reasons seem to be as follow :—

1. To eliminate structural failures.
2. To reduce the weight of the structure.
3. To reduce design time.
4. To reduce fabrication time (and cost).
5. To improve the service and maintenance characteristics of the aeroplane.

Items 1 and 2 should actually be combined under the customary term " strength/weight," as there would be no danger of structural failure if the designer were allowed to use an unlimited amount of weight.

Item 3, the reduction of design time, is particularly important at the present moment. Anything that can be done to speed up the structural analysis, and to make it more reliable, will have an important effect in reducing the time required to release the drawings.

Item 4, reduction of fabrication time, is also extremely important under war conditions. The modern philosophy of design or production is to consider the final manufacturing problems from the very beginning of the design.

Item 5, service and maintenance, has received too little attention in the past. More accurately, this phase of design has not been suitably correlated with other phases, and has usually had to be taken care of after the aeroplane has been put in service. The ideal research and development programme will consider all of these items and will give them proper weight from the beginning.

Classification of Structural Research Problems.

Having classified the objectives toward which structural research should be aimed, it is now advisable to subdivide the research problems themselves on some suitable basis. Table I shows how the subjects are broken down for greater convenience.

Considering the three subdivisions shown in Table I, we probably know less about external loads than any other phase of aeroplane design. This condition will, no doubt, always exist, as it is impossible to determine exactly the loads which will act on a certain aeroplane. However, it is possible to obtain statistics on the most probable loads that will be imposed on a type of aeroplane, which information is extremely valuable when dealing with the problem as a whole.

TABLE I. CLASSIFICATION OF STRUCTURAL SUBJECTS.

1. External Loads :—
   - (*a*) Flight :—
     - Manœuvring
     - Gusts
     - Control surfaces
     - Engine
   - (*b*) Ground (or water) :—
     - Landing
     - Taxiing
     - Handling (towing)
   - (*c*) Miscellaneous :—
     - Local (inertia) loads
     - Controls (pilot)
     - Pressure (cabin, hydraulic, etc.)
     - Rigging and service
     - Flutter
     - Compressibility effects

2. Internal Loads :— (Stress distribution)
   - (*a*) Framework :—
     - Trusses
     - Space frameworks
     - Rings
     - Relaxation methods
   - (*b*) Beams :—
     - Solid type
     - Box beams
     - Shear lag
     - Unit method
     - Torsion
     - Cutouts

3. Allowable (Failing) Loads :—
   - (*a*) Buckling (instability) :—
     - Columns
     - Panels
     - Effective width
     - End fixity
     - Edge restraint
     - Shear beams
     - General instability
   - (*b*) Failure of materials :—
     - Combined loads
     - Interaction curves
     - Stress concentration
     - Fatigue
     - Impact
     - Joints
     - Structural indexes
     - Material properties

Once we realize that the design of the structure starts out on a probability basis, we can guide ourselves accordingly in dealing with other phases of the problem which are more susceptible to exact analysis.

Thus the first suggestion to be offered for future research is a comprehensive study of the probable errors or deviations entering into the design and fabrication of an aeroplane structure. To make such a study, it would be necessary to know or to estimate the probable errors or deviations for such things as load factors, mathematical calculation, material properties, manufacturing tolerances, and all the other factors which have any appreciable effect on the final result.

Keeping in mind the ultimate yardstick of weight reduction, the suggested study could be expanded to include estimates of the weight involved for each factor—that is, the weight that would have to be added to reduce the particular probable error to zero or some arbitrarily small value. This would reveal cases in which a small amount of additional weight would more than compensate for a lack of accurate data, for short-cut methods of analysis, for lack of quality control in manufacturing, etc.

The ultimate form of such an investigation would consist of a summation of all possible causes of structural failures, the final result being a certain small probability of failure which is considered to be tolerable. In making this summation, it would, of course, be essential to combine various probabilities in the proper manner. For instance, if a certain failure could result only through simultaneous occurrence of several unfavourable errors or tolerances, the probability of such failures would be much less than in the case where any one error would cause failure independently of the others. A clear understanding and evaluation of these principles is of great importance not only to the designers, but particularly to government agencies responsible for the airworthiness requirements and specifications.

* Numbers in parentheses refer to the Bibliography at the end of the paper.

External Loads.

Since this phase of aircraft structural research does not have a direct counterpart in other engineering fields, it will not be taken up in detail in this paper. Furthermore, the problems of external loads are, in general, well understood by aircraft engineers and by various government research agencies, particularly the N.A.C.A., which has done a great deal of work on the subject. There are, however, a few points which may be of interest to non-aircraft people, and upon which the experience of other industries may be beneficial to the aeroplane designers. One of these pertains to the instruments and methods used in collecting statistical data from which the external loads may be determined. As previously noted, it is practically impossible to calculate accurately the loads to which the aeroplane will be subjected. A statistical approach to this problem must, therefore, be used.

The best example of what has already been done in this field is the instrument known as the V-g recorder, a description of which is given by R. V. Rhode (2). This instrument was developed by the N.A.C.A. and is now in constant use. The records obtained on various types of aeroplanes under different flight conditions have furnished a most valuable background for the determination of the flight loading conditions.

The instrument furnishes a plot of maximum acceleration (normal to the wings) against the air speed (dynamic pressure) of the aeroplane. The combination of these two factors not only enables us to calculate the total load acting on the wings, but also gives us a basis for determining the distribution of these loads.

Our greatest need at the present time is for more statistical data on local accelerations or stresses in various parts of the aeroplane structure, particularly the tail surfaces and the landing gear. It is desirable to know, not only the maximum accelerations or stresses experienced over a long period of time, but also the number of times that various values of these factors have been experienced. This type of information is of great value in connection with fatigue conditions, for which relatively little information is available on the aeroplane.

Although accelerometers have long been used as a method of obtaining such data, it now appears that the best instrument for obtaining local information is the electric strain gauge. The technical difficulties in obtaining suitable accelerometers are not the only reasons for this statement, since it should be remembered that acceleration readings alone are of no particular interest, as they must be multiplied by an effective mass before the loads and stresses can be calculated. The problem of determining the proper effective mass often turns out to be more difficult than that of obtaining the local acceleration. The use of the resistance type of electrical strain gauge permits us to record the strains at any desired point on the structure. This type of strain gauge, which is nothing more than several loops of fine wire pasted to the structure, is described by C. E. Strang (3). The gauge is easy to install, takes up very little space, and enables a large number of readings to be made relatively quickly. The ideal solution to the external load problem would be to have a large number of such gauges installed at strategic points on the structure of the aeroplane and to devise some means of recording statistically the frequency distribution of the strains developed in actual operation of the aeroplane. Such equipment would have to be relatively inexpensive and foolproof, so that it could be installed in a large number of aeroplanes and left unattended for relatively long periods. This problem is being worked on by the N.A.C.A., Civil Aeronautics Authority (C.A.A.), and other agencies.

The de Forest type of scratch gauge (4) is quite useful in obtaining statistical information on the strains developed in the structure, but it has the disadvantage of using up the record in a relatively short time. Some attempts have been made to modify this gauge so that it would not operate until after certain strain has been exceeded. This would eliminate a large portion of the record, in which we are not particularly interested, and would thereby extend the time-range much farther. Fig. 1 shows the ordinary scratch gauge; other types are available for various purposes.

Dr. Pugsley (1) stated the case very clearly when he said:—

". . . A second factor for comment is the part which statistical methods and view-points may play in the future. It would seem to behove research workers to make themselves at home with the ideas of statistical theory, to acquire a detailed knowledge of existing structural statistics, and to interest themselves in the collection of further data."

Although the greatest need appears to be for additional statistical information, much remains to be done on the mathematical analysis of the aeroplane as a flexible body subjected to various loading conditions. Most of the work in this field has been done in connection with gusts (5). There have also been some investigations for landing conditions. The greatest need at the moment appears to be for a more complete and rational analysis of the behaviour of the aeroplane under various impact conditions and for taxiing over rough ground. Such a study should include the effects of structural flexibility, shock-absorber characteristics, and aerodynamic effects. Dr. E. G. Keller, consulting engineer for Lockheed Aircraft Corporation, has been working on this problem recently, and the preliminary results of his investigation have proved to be of considerable interest.

These results indicate that the effects of structural flexibility are not of great importance in the case of a hard landing, in which the load is applied over an appreciable interval. On the other hand, in taxiing over rough ground, the forces are applied to the landing gear quite rapidly, and it is entirely possible that the loads developed under such conditions are much greater near the point of contact than at some distance away from this point. There are also indications that, in taxiing over rough ground, the forced oscillation of the tail surfaces may be further built up by aerodynamic effects, which may account for the high values of local acceleration that have been measured at the tail surfaces of actual aeroplanes. The importance of this particular subject is emphasized by the fact that there have been several cases of failure of tail surfaces under such conditions, particularly for the nose-wheel type of aeroplane to which most of these remarks apply.

Remembering that weight reduction is the ultimate objective of all structural research, it should be noted that further research along these lines will tend to eliminate weight by telling us how to distribute the structural weight more efficiently. For instance, it is useless to provide a great deal of weight for the main landing gear if the nose gear is relatively under strength and would always fail first. In general, it appears that further weight savings might be effected by providing more shock absorption for landing and taxiing, as this will reduce the magnitude of the loads for which a large part of the aeroplane structure must be designed.

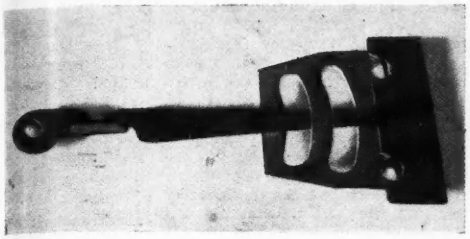


Fig. 1. Scratch type of Strain Gauge.

Flutter and Compressibility.

The problem of flutter is rather difficult to classify. Although structural theories and measurements are involved, the problem appears to be more aerodynamic and mechanical in nature. Structural theory assists by predicting the stiffness of the structure, but a more direct approach is to measure the natural frequencies by vibration tests while the aeroplane is on the ground. Mechanical engineers should be interested in the possibilities of flutter-prevention devices. If such devices could be made light enough they could replace present methods of balancing movable surfaces and thereby save a considerable amount of weight. Ordinary "dashpot" devices cannot usually be used, because of the complexity of the flutter modes and the danger of interfering with free movement of the control surfaces. The difficulty of preventing flutter becomes greater as the speed of the aeroplane is increased. Flutter research must be vigorously pursued on all fronts, in spite of the great advances that have been made in this field during the past few years.

Compressibility is rapidly becoming an important aerodynamic problem for all high-speed aeroplanes. The modern trend toward high-altitude flying and fighting has brought us rapidly closer to the compressibility problem. The structural engineer is interested primarily in its effect on the distribution of air loads on the structure and in the increased values of local air pressures.

Internal and Allowable Loads.

After the external loads acting on the aeroplane have been determined, it is necessary to find out how these loads are distributed through the structure. The mass distribution of the aeroplane determines where the loads go, but it is the stress analyst's job to find out how they get there. The methods of analysis developed for other types of structures are generally applicable to the aeroplane, with a few exceptions and refinements which will be discussed.

In dealing with different structural components it is convenient to combine the subjects of internal and allowable loads, although they are kept fairly well separated in an actual analysis. For the present purpose of discussing research problems the type of structure will be used as a basis and no effort will be made to distinguish clearly between internal and allowable loads.

Trusses and space frameworks are quite adequately covered by available literature and data. In recent years, attention has been directed mainly toward the problems of end fixity and moment distribution, in an effort to obtain a more accurate understanding of the interaction which results from "continuous" joints.

Relaxation methods, originally introduced in this country by Prof. Hardy Cross (6) and highly developed by Professor Southwell (7), have proved to be most useful, particularly for continuous beams having several supports. Modern aeroplanes usually have a relatively small amount of structure which falls in this category, because of the almost universal use of the shell and plate types of structures (metal skin). Hence, it would seem wasteful to devote much research effort to the framework, the continuous beam, and the beam column, in so far as the aeroplane is concerned.

Rings and frames continue to be of special interest to aeroplane designers, particularly at the juncture of shell structures (wing to fuselage, for example). To be of use, methods of analysis must be capable of handling any arbitrary shape, any loading condition, and should permit consideration of stresses beyond the elastic limit. Too much emphasis has been placed on the academic side of this problem. The present state of knowledge is fairly complete, however, and the only remaining need is for a general "cleaning up" and organization of all the best information, including test data.

The solid beam is usually treated by the classical beam theory ($F = My/I$), which assumes a linear variation of strain over the cross section and a constant ratio of stress to strain. In aeroplane work, where weight-saving is so important, it has been necessary to employ refinements, particularly because of the use of new materials having different types of stress-strain diagrams. The variety of cross-sectional shapes used also complicates the problem. The theory of bending in the plastic range is not all new, but it seems to have been greatly overlooked in practical application.

At Lockheed we have recently completed a small series of tests of a family of shapes, to determine a suitable method of predicting the failing load bending. Fig. 2 shows some of the specimens tested. By utilizing the stress-strain diagram of the material it has been possible to correlate the bending theory with actual test results. Charts have been derived from which the ultimate and "yield" bending moments may be determined for all the materials commonly used for casting, forgings, and machined parts. A paper* is in process covering the entire developments, which was carried out by the Stress and Structures Research

* "Bending Strength in the Plastic Range," by F. P. Cozzone, presented at Annual Meeting of the Institute of Aeronautical Sciences, January, 1943.

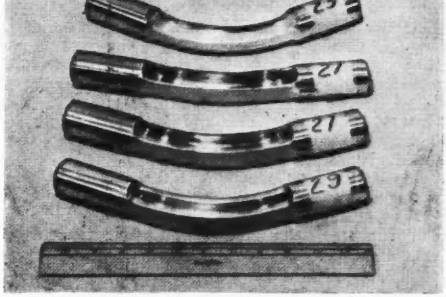


Fig. 2. Typical Specimens for Bending Tests.

Groups at Lockheed. Much work remains to be done to extend present knowledge into the fields of torsion, combined axial load and bending, combined torsion and bending, and various cross-sectional shapes. In the thinner sections the effects of local buckling and instability will have to be considered. We must also be able to predict the loading which will produce the first permanent set, as well as the ultimate load.

Shells.

The shell type of beam is widely used in modern aeroplanes, particularly for fuselages and wings. In determining the axial stresses due to bending, we find that the classical beam theory again has serious shortcomings. Under pure bending (no shear) the assumption of plane cross sections is valid, provided that the bending moment is applied in accordance with the theory (i.e., linear strain variation), or, alternatively, that the cross section is far enough away from the point of application so that end effects can be neglected.

However, when the bending is the result of a shear load which is being transmitted by the structure, we are no longer safe in assuming that plane cross sections remain plane. In the case of a wing shell such as sketched in Fig. 3, for instance, the increments of axial loads in the top and bottom, as we move from tip to root, are applied along the lines of attachment of the "vertical" shear webs and must travel across the "horizontal" flange material. This relatively long path permits appreciable shear deflections which cause the cross section to distort from the assumed plane section. The result is an increase in axial stress in the flange material near the webs, and a decrease in that portion farthest away from the webs, as compared with the results of the classical beam theory.

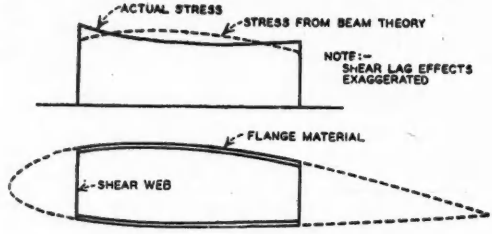


Fig. 3. Shear-Lag Effect in Wing Beam.

This so-called "shear-lag" effect has received much attention in recent years, and methods of handling it in stress work are now available. One important point that has received too little attention is the effect of the geometry of the structure, particularly the effects of taper in depth and width. It can easily be seen that if the beam flanges were made wider from tip to root, at such a rate that the new flange material added just took care of the new axial loads added by the webs, there would be no "shear-lag." Most wing beams tend to approximate this condition; in fact, the optimum design for maximum efficiency would give just such a beam.

As shown by Cozzone and the author (8), the use of the classical beam theory and its companion shear formula (shear flow = SQ/I) gives a misleading picture. The "unit method" of analysis was therefore adopted, to take care of the effects of taper in depth and plan form. This method also reveals the effects of abrupt changes in distribution of flange material, such as found at cutouts.

The problem of shear lag may become important in some cases, but, in general, is felt to be less critical, from a weight and safety standpoint, than that of cutouts, to be discussed later.

Actual use of the "unit" method of beam analysis, as described elsewhere (8) and (9), has shown the need for a "three-dimensional" beam theory. For instance, it is not uncommon to find that the neutral plane for an actual wing structure has "steps" in it, as well as abrupt "twists." These, of course, indicate that the beam material has not been distributed as uniformly as might be desired. (If the discontinuities are too bad, the design should be changed, but this cannot be carried to an extreme.) It does not seem possible for the neutral plane to have abrupt discontinuities of this nature. We have tried "fairing" it in some arbitrary manner, and have found that this greatly improves the flow of axial loads and reduces the local shear flows. Strain-energy checks also show that the faired neutral plane involves less strain energy, hence is closer to true conditions. Without some such modification, the "unit" method will indicate fictitiously high local shear flows wherever there is a slight disturbance in the continuity of the structure. The proper type of modification to use is a research problem for the future, and is closely connected with the cutout problem. In any case, "spanwise" variations must be brought into the picture in some manner.

The shell type of beam usually must resist some torsion, in addition to the direct shear and bending loads which it carries. In analyzing a shell for torsion, it is customary to use the simple shear-flow formula $q = T/2A$, where q is the shear flow, T is the torque, and A is the area enclosed by the median line of the shell skin. In contrast to the beam theory, this formula is based on the assumption that the cross section is free to warp. This causes something of a dilemma, as the same structure obviously cannot behave in two different ways.

Fortunately, the same tendency to warp which caused the beam theory to become inaccurate, also tends to make the shear-flow formula more nearly correct. It has been proved by theory and tests, for instance (10), that the errors are greatest at the point where the structure is forced to maintain a plane cross section. The effect dies away quite rapidly and becomes negligible at some distance away from the point where the cross section is forced to remain plane.

In an aeroplane wing symmetrically loaded, the plane of symmetry acts like a rigid plane bulkhead. Hence, the torsional-restraint effect will be a maximum at the root of the wing. This effect not only changes the shear flow around the shell, but also introduces additional axial stresses in the flanges and webs. Some of these will add to the axial stresses produced by bending, while others will subtract.

The subject of end restraint in torsion is now fairly well understood. Considerably more work needs to be done, however, on the problem on introducing a large "local" torque at some intermediate point on the shell structure, such as at the nacelle and landing-gear attachments.

Cutouts.

Cutouts in shell structures represent one of the meanest of problems facing the aeroplane designer. There has probably never been an aeroplane in which cutouts of some sort were not present. It is obvious that a system of analysis for shell structure will be of little use unless it can be extended to include the effects of cutouts.

In treating cutouts, it is convenient to consider separately the effects of axial and shear loads. The

usual approach in the case of a small cutout is to ignore it in the original analysis and then treat it as a local problem. The structure can be presented by a number of "flange units" each of which is carrying an axial load, as shown in Fig. 4. After the material is cut away, the loads in the cut elements must be diverted around the cutout. This introduces three problems: (*a*) additional material must be provided along the edges, to avoid overloading the original material; (*b*) this material must be extended beyond the cutout far enough to permit it to pick up the loads from the cut elements; (*c*) additional shear material may have to be provided to take care of the local shear flow caused by the rapid transfer of axial load. These three problems are obviously interrelated in a very complex manner.

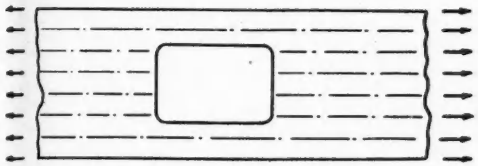


Fig. 4. Cutout under axial loading.

Fig. 5. Lower wing cutout.

The key to the analysis and design of a cutout of this type is a knowledge of the manner in which the axial load "decays" in an element which has been abruptly or gradually cut off. This will depend, of course, upon the stiffness of the surrounding material, the distance through which the load must be transferred (laterally), and several other secondary variables. Many simple tests could be made to obtain basic data on this important problem.

In many cases the cutout is surrounded by a frame, which may be relatively stiff against bending in the plane of the shell. In such cases, the frame may play an important part in the transfer of load and will be subjected to high bending stresses, particularly at the corners. It is also common practice to add an extra plate or "doubler" all around the cutout. This helps carry both the axial and shear loads. If the cutout does not occur in a highly-stressed portion of the structure, this simple treatment may suffice.

A very important type of cutout is often found in the bottom of a wing structure. Sometimes the entire bottom is cut away between the shear webs, leaving a structure such as shown in Fig. 5. All the lower flange material must be concentrated at the shear webs. This type of cutout is particularly difficult to analyze in torsion and has a serious effect on torsional rigidity if it extends over some distance.

The effect of a small cutout in a shell carrying torsion (or pure shear) is somewhat similar to its effect under axial loads, except that two axial fields now exist, i.e., tension and compression at 90 deg. (In some cases the tension field will predominate, due to buckling under the compressive forces.)

If the cutout is small, a heavy frame can be used to replace the material removed, particularly in the case of round or oval cutouts. At present there is very little information to guide the designer in his efforts to do the job with minimum weight.

If the "shear" cutout is large, a frame may become impractical as the method of reinforcement. It is then necessary to divert the shear flow into the structure surrounding the cutout. This requires, at the ends of the cutout, bulkheads or ribs capable of redistributing the shear flow. An elementary attempt to analyze such a case may be made by using the shear flow formula ($q = T/2A$) and applying the beam theory over the portion containing the cutout. Test data indicate that large errors may be introduced by such methods because of the rapidly changing local conditions.

One reason for the lack of data and literature on such subjects may be that they are most difficult to handle by formal elastic theory and mathematics. A semi-empirical approach seems to be necessary, in which empirical (test) data on load transfer or decay are used in conjunction with the basic conditions of equilibrium and continuity.

Buckling—Instability.

Before proceeding further with the major problems of shell analysis, it is necessary to discuss the important problem of instability. We can practically omit the simple column from this discussion, as structural literature abounds with column theories and experiments. At the present time, the aeroplane designer uses the charts in ANC-5 (11) for steel- and aluminium-alloy tubing. The "tangent-modulus" theory has been shown to be quite accurate as a basis for the prediction of column curves for any new material, provided that the compressive stress-strain diagram is available (12, 13). Professor Van den Broek has recently written an interesting paper (14), in which he advocates the eccentric-column theory as a basis for column analysis. There is, of course, no argument about the effect of initial eccentricity, but it would seem that our present knowledge of ordinary column failure is so far beyond our knowledge of other structural problems that the subject should be closed "for the duration."

The buckling of flat plates has likewise received much attention and the theory is well worked out (15). Complete data are lacking, however, on the conditions which cause permanent buckles. In most stiffened shell structures, the sheet may buckle before the ultimate load is reached. The usual buckling theories give only the load at which buckling begins. If the sheet is restrained by stiffeners it will usually be impossible even to detect any buckling at the theoretical load.

In modern aeroplane work, there are actually three different criteria to meet:—

(*a*) No visible buckling in normal flight (typical load-factor range: 0.175g to 1.25).

(*b*) No permanent wrinkling at "limit" load (varies from 3g to 8g, approximately, depending upon type of aeroplane).

(*c*) No failure below required "ultimate" load (usually 1.5 times "limit" load, i.e., 4.5g to 12g).

Part (*a*) is covered fairly well by the existing buckling theories. Part (*b*) is more difficult and there are relatively few data on it (16). Part (*c*) involves the failing of the stiffeners, and also the problem of "effective width."

Much work has been done on "the effective-width" problem (17). In fact, the author is inclined to believe that too much time has been spent on the theory, as it is relatively easy to test a small panel representing the structure in question. Such a test takes care of many troublesome factors, such as loca buckling stringers, inter-rivet buckling of sheet, twisting of stiffeners, effect of sheet wrinkles on

stiffener strength, etc. At Lockheed we have made attempts to correlate all our panel test data by means of existing theories, but up to this time we still find it advisable to make a few panel tests whenever we depart very far from combinations already tested.

While on the subject of panel testing, it is recommended that more attention be given to end fixity, as affected by various factors present on the aeroplane. There is much confusion and argument on this subject. Some of the typical factors involved are :—

(*a*) Different adjacent panel lengths.
(*b*) Restraint from ribs and bulkheads.
(*c*) Effects of lateral loading (air loads, water loads, etc.).
(*d*) Influence on bending curvature (as in a wing under bending loads).
(*e*) Plastic (short column) range.

Stewart Way cleared up many of these points in an article in 1935 (18). Other valuable work has been done since, but not in proportion to the importance of the problem. The coefficient of fixity may actually vary between 1 (or even less) and 4. The arbitary selection of a value of 1.5, for instance, would represent a 50% increase in strength over a pin-ended column (in Euler range). How many other cases can be found in which the designer knows so little about such powerful variables ? Considering that the largest single share of structural weight is involved in such structure, we cannot afford to be too " conservative " in order to cover our lack of knowledge.

Another effect that should yield valuable weight-saving results is that of edge restraint. In wings, for instance, the four corners of a single-cell box beam are often heavy " cap-strips " which are completely supported against column failure. Further investigation of this case may be in order, as there is very little information on the minimum size of cap strip required. To some extent, the problem is one of combined loadings, as the corner cap is loaded in compression and also acts as a shear-transmitting member. This would be specially severe if the shear web were designed as a tension-field beam, which would apply additional bending loads to the cap strip

Shear beams of the plate or thin-web type have been thoroughly investigated, and the designer now has fairly adequate analysis methods and data. At present there is a wide variation in ideas on this subject, part of which is due to the lack of a single co-ordinated source of information.

It should be possible to determine the best arrangement of material for different loading intensities, using a " structural index " such as Wagner's $\sqrt{S/h}$ or the square of this, S/h^2, where S is the shear load and h the depth of the beam. (Such indexes will be discussed more fully later in this paper.) It will be found, for instance, that the truss type is lightest in the lower range, the thin-web type in the intermediate range, and the heavy plate girder or corrugated web in the heavy loading range. Various companies have made studies of this sort, but it is not likely that any of them has been able to assemble and digest all of the available test data on the subject.

The shear beam which tapers in depth is of special interest to the aeroplane designer. This subject was relatively neglected for a long time, but it is encouraging to note an increased number of papers in this field. Too many of these are of the " classical " or academic type, however, and are of little immediate use to the designer.

Curved shear beams are widely used in aeroplanes, as in fuselages, nacelles, and " D-tube " types of wings. Much less inforamtion exists than in the case of flat beams. If the sheet is sharply curved, it will resist high shear loads before buckling, but will tend to form a permanent wrinkle when it does buckle. The slightly-curved thin-web beam, on the other hand, behaves more like the flat beam, the wrinkles forming early and disappearing when the load is removed. Stiffener spacing and strength play an important part.

It should be noted that the laboratory testing of shear beams, particularly curved ones, presents a serious problem. It is very difficult to reproduce, in a jig, the exact conditions which would exist in the aeroplane.

Combined Loads.

Combined loads are always encountered in aeroplane structures. The interaction curve, or stress-ration method (19), has been found quite valuable as a means of bridging the gap between the different loading conditions. An interaction curve is sketched in Fig. 6, indicating how it is used to handle combined axial and shear loading. Terms P*cr* and S*cr* are the allowable (critical) loads for axial and shear loading, respectively, while P and S are the actual applied loads. The dotted lines indicate one particular combination of loads.

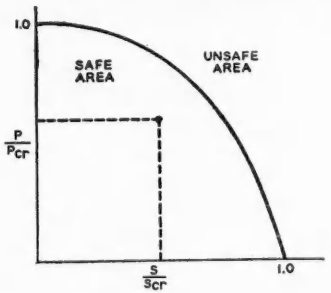


Fig. 6

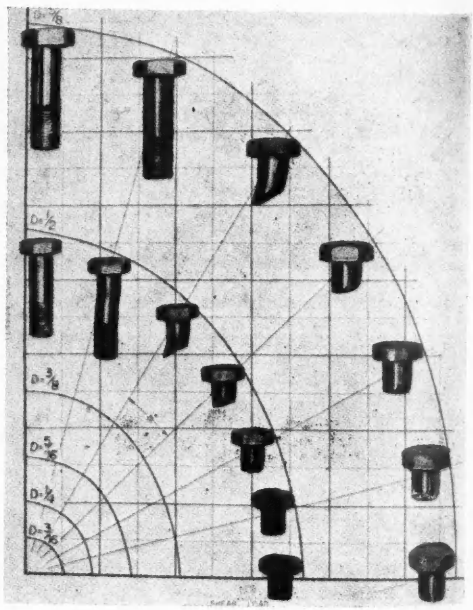


Fig. 7. Failure of Bolts under combined shear and tension.

This type of curve has the advantage of always being correct at either end; thus, if one loading condition predominates, the error caused by a poor assumption for the interaction curve cannot be large. A straight line between the two end points will be safe in practically any case except one in which secondary effects occur (such as combined axial and bending loads where the axial loads increase the bending loads).

The interaction-curve method eliminates the need for a detailed investigation of each case, a point which is of great value in actual stress work. Fig. 7 shows how this idea has been applied in the case of bolts loaded in combined tension and shear. (Lockheed test data.)

General instability is important phenomenon in which buckling failure occurs over a relatively large area of the shell, including both longitudinal and transverse stiffeners. This subject has been under investigation by the California Institute of Technology for the past several years and valuable results have been obtained. The subject is of particular importance in the case of very large aeroplanes, as it would be inefficient to use frames or bulkheads any heavier than necessary. The general instability problem is a typical example of a structural theory which was practically unheard of until the thin shell structure of the aeroplane necessitated its development. Some of the earlier work on this subject has been described (20, 21).

Before leaving the subject of shell structures, it might be stated that the present trend of design appears to be toward the double-skin type of construction. This may take the form of the "cardboard box" (Fig. 8), the "separator" type (Fig. 9), the "filler" type, or the "inner stamping" (Fig. 10). Most of these ideas have actually been used in Lockheed aeroplanes. The double skin represents the most efficient use of material in bending and shear, as compared with sheet-stringer construction.

(*To be continued*)

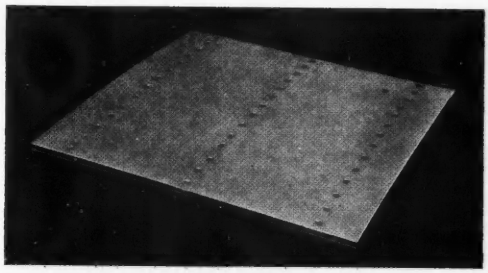


Fig. 9. Spacer Type of Double-Skin panel.

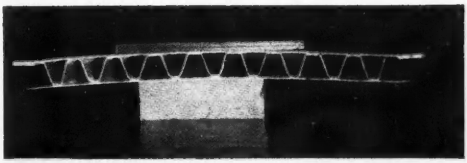


Fig. 8. "Cardboard-Box" Type of Double-Skin Panel.

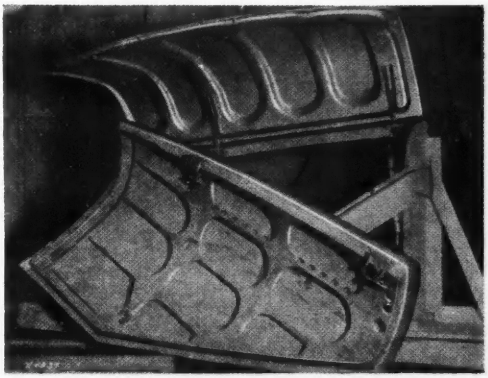


Fig. 10. Use of Inner Stamping.

A HYDRAULIC RAM WORKING ON WAVE PROPAGATION

Invented by Prof. Dr. G. EICHELBERG, of the Federal Technical College, Zurich. (From *Schweizerische Bauzeitung*, Vol. 120, No. 17, 24th October, 1942; pp. 191-193.)

Introduction.

WHEN a pipe filled with water at pressure p_2 is suddenly connected to a tank containing water at pressure p_1, a wave of pressure is propagated along the pipe with the velocity of sound, C_s (Fig. 1), and, as a result of the compression, there is influx to the pipe with velocity v. Reflection takes place at the far end and the wave travels back to the tank, raising the pipe pressure still further by an amount $p_1 - p_2$ above the pressure p_1 then prevailing. The reflected wave stops the flow caused by the initial wave, and, at the instant of reaching the valve, it has brought all the water to rest. If the valve is closed at this instant, the pressure in the pipe stays at the value $2p_1 - p_2$.

If the pressure p_1 in the tank is less than p_2 in the pipe, the process is similar, but the pressure rise is negative, and a wave of reduced pressure is propagated which, on reflection, causes a further reduction amounting to $p_2 - p_1$. The water in this case expands from the pipe into the tank with velocity v (Fig. 2).

The compression wave causes the pressure in the pipe to exceed, and the expansion wave causes it to fall below, the pressure in the tank; so that, in the one case, the pipe can deliver to a higher lever, but, in the other case, only to a lower level.

This led Prof. Eichelberg to the novel idea of distributing water at high and low pressures from a source of intermediate pressure through the continuous operation of waves of compression and expansion, and so to the invention of a new device for lifting water without the need for turbine and pump.

Method.

In Scheme I (Fig. 3) the action of the control valves, A and B, falls into a cycle of three phases, which are shown in Fig. 4.

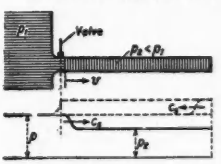


Fig. 1. Wave in positive pressure in pipe.

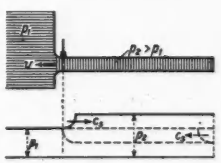


Fig. 2. Wave of negative pressure.

(C_s is the velocity of wave propagation in this and other figures, but is replaced in the text by the symbol *a*.)

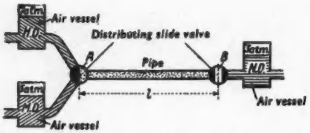


Fig. 3. Diagram of Ram. H.D. = High pressure, M.D. = Intermediate pressure, N.D. = Low pressure.

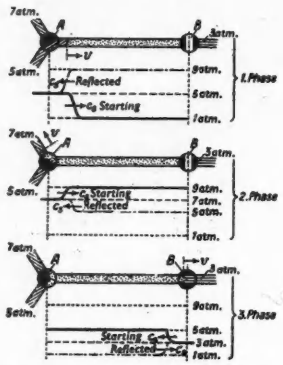


Fig. 4. Cycle of operations—Scheme I.

Initially, the pipe, at a pressure of 1 atmos., is suddenly connected to the air vessel M.D., which is at the intermediate pressure of 5 atmos. A wave of pressure is propagated along the pipe with the velocity of sound, is reflected at B, and returns to A, raising the pipe pressure to 9 atmos. During this phase, water flows from the I.P. air vessel, M.D., into the pipe. At the instant when the wave returns to A, valve A is switched over to the high-pressure air vessel, H.D., thus beginning the second phase. Since the H.P. air vessel, H.D., is below pipe pressure, a wave of expansion then travels along the pipe and after reflection comes back to A at 5 atmos.

During this phase a certain amount of water passes from the pipe into the H.P. vessel, H.D. The second phase ends with the closing of valve A as soon as the wave of reduced pressure reaches it. The water is now at rest at 5 atmos. The opening of valve B starts the third phase in which a wave of expansion runs along the pipe, returning to B at 1 atmos., and flow takes place from the pipe into the low-pressure air vessel, N.D. Valve B is now closed and the initial conditions are restored. The volume of water which flowed into the pipe from the I.P. vessel, M.D., by reason of the compression from 1 atmos. to 9 atmos., has been discharged, one half into the H.P. vessel through expansion from 9 to 5 atmos., the other half into the L.P. vessel through the further reduction in pressure from 5 to 1 atmos.

Theory.

The length of the pipe being 1 metre, the distance travelled by the waves at the velocity of sound, a metre per second, is 61 metres in each complete cycle. The frequency is, therefore, $n = \frac{a}{61}$ cycles per second.

Let F be the cross-sectional area of the pipe in square metres and V, = lF cubic metres, the volume of water in the pipe at the start of the first phase.

At the end of this phase the volume is compressed by the amount ΔV, given by

$$\frac{\Delta V}{V} = \frac{\Delta p}{\rho a^2}$$

where Δp is the pressure rise in kgs. per sq. metre. = (9 — 1) 10,000 = 80,000 kg./sq. m. in this example. (*Note.*—The atmosphere is commonly taken as 1 kg./sq. cm.) and ρ is the density in $\frac{kg \cdot sec^2}{m^4}$ $(= \frac{1}{g} \times$ weight per unit volume).

The volume ΔV is filled from the I.P. vessel, M.D., and this is the quantity taken from it in each cycle. The delivery Q per second is, therefore,

$$Q = n\,\Delta V = \frac{a}{61} \times 1F\,\frac{\Delta p}{a^2} = k\Delta p \text{ cub. m./sec.} \quad .. \quad (1)$$

where $k = \frac{F}{6\,\rho a}$, and Δp = p max. — p min.

(p max. and p min. are respectively the pipe pressures at the end and at the beginning of the first phase).

Let the intermediate pressure be $p_M = \frac{1}{2}$ (p min. + p max.)

„ high „ p_H

„ low „ p_N

and let the delivery per second to the H.P. and L.P. air vessels be respectively Q_H and Q_N.

Then $Q = Q_N + Q_H$

and, since $p_H = \frac{1}{2}(p_M + p_{max})$ in this example, it follows that $Q_N = Q_H$.

It will now be shown that, with given values of p_M and Δp, there is in general a definite pressure p_N, and therefore a definite pressure p_H, for any ratio

$$\nu = \frac{Q_N}{Q_H} \quad .. \quad .. \quad .. \quad (2)$$

The relations between the pressures are shown in Fig. 5, $\bar{p}$ being the pressure at the end of the second phase.

Since $p_H = \frac{1}{2}(\bar{p} + p_{max})$ and $p_N = \frac{1}{2}(p_{min} + \bar{p})$

then $p_H - p_N = \frac{\Delta p}{2} = p_M - p_{min} = p_{max} - p_M$ .. (3)

Conforming to equation (1),

$Q_N = k(\bar{p} - p_{min}) = 2k(p_N - p_{min})$.

$Q_H = k(p_{max} - \bar{p}) = 2k(p_{max} - p_H)$

or, putting $\Delta p_H = p_H - p_M$, and $\Delta p_N = p_M - p_N$,

$Q_N = 2k\,\Delta p_H$, and $Q_H = 2k\,\Delta p_N$ .. .. (4)

From (4) .. $\nu = \frac{\Delta p_H}{\Delta p_N} = \frac{p_H - p_M}{p_M - p_N} = \frac{p_N - p_{min}}{p_{max} - p_H}$

or $\nu\,p_H + p_N = p_{min} + \nu\,p_{max}$

This, with equation (3), gives

$$p_H = p_M + \frac{\nu}{1+\nu} \times \frac{\Delta p}{2} = p_M + \frac{Q_N}{Q} \times \frac{\Delta p}{2}$$

$$p_N = p_M - \frac{1}{1+\nu} \times \frac{\Delta p}{2} = p_M - \frac{Q_H}{Q} \times \frac{\Delta p}{2} \quad .. \quad (5)$$

The energy balance follows from (4) and (1)—and is

$$Q_H\,\Delta p_H = Q_N\,\Delta p_N = \frac{Q_N\,Q_H}{Q} \times \frac{\Delta p}{2} \quad .. \quad .. \quad (6)$$

For any installation the intermediate pressure p_M is given. The pressure p_{min} should not be less than 1 atmos. in order to avoid cavitation and leakage of air into the system. p_M and p_{min} determine $\Delta p = 2(p_M - p_{min})$, and together with ν in equations (5) or Fig. 6, they also determine p_H and p_N.

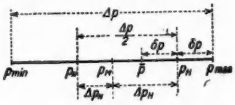


Fig. 5. Diagram of pressures. Scheme I.

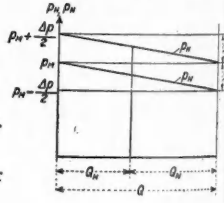


Fig. 6. Pressures and water quantities.

Starting Up.

If an air compressor is available the air vessels can be brought up to their working pressures before putting the system into operation; and delivery begins as soon as the speed of the rotary valves corresponds to the frequency of the system. But, even if there is no supply

of compressed air, the working pressures can still be obtained from the inlet pressure p_M by the building up of pressure waves.

Let the H.P. air vessel be originally at the inlet pressure p_M of the I.P. air vessel, and let the pipe and the L.P. air vessel be at pressure p_N corresponding to v. By operating the system with the phase sequence previously described, the to-and-fro motion of the pressure waves delivers water to the H.P. and L.P. vessels, alternately. While the latter is kept at constant pressure, p_N, say by a spring-loaded valve, the pressure in the H.P. vessel at the beginning of any cycle is, in consequence, higher, and the pipe pressure lower, than at the beginning of the previous cycle. With each successive approach to the value p_H corresponding to v, the waves of pressure and the quantities delivered come nearer to their steady operating values.

Efficiency.

The actual quantities delivered to the H.P. and L.P. vessels differ from the theoretical values because of the imperfect tightness and the overlapping of the opening times of the control valves; experiments show that the difference ΔQ is the same for both vessels. The actual values are

$$Q_H\text{eff} = Q_H - \Delta Q \quad \text{and} \quad Q_N\text{eff} = Q_N + \Delta Q$$

The hydraulic efficiency is defined as the ratio between the energy given up to the energy put in. Theoretically this ratio is $Q_H \Delta p_H$ divided by $Q_N \Delta p_N$, which, by (6), is unity; but actually

$$\eta_h = \frac{Q_H\text{eff}\,(p_H - p_M)}{Q_N\text{eff}\,(p_M - p_N)} = \frac{Q_H\text{eff}}{Q_H} \times \frac{Q_N}{Q_N\text{eff}} = \frac{1-\frac{\Delta Q}{Q_H}}{1+\frac{\Delta Q}{Q_N}} \quad .. \quad (7)$$

From this it follows that, for good efficiency, $\frac{\Delta Q}{Q_H}$ and $\frac{\Delta Q}{Q_N}$ must be small, which is possible only if Q is comparatively large. The power L required to operate the control valves must be allowed for in determining the overall efficiency, which is

$$\eta_{tot} = \frac{Q_H\ \text{eff}\,(p_H - p_M)}{Q_N\ \text{eff}\,(p_M - p_N) + L} \quad .. \quad .. \quad (7a)$$

In addition to losses due to leakage at the valves, there are pressure losses due to throttling by the valve passages. These can be reduced to a minimum with properly designed valves. Air in the piping has a particularly detrimental effect by retarding the oscillations, flattening the wave front and diminishing the quantity delivered.

Increase of Output by Scheme II.

The losses due to leakage are constant for any given pressures and valves. Therefore, the delivery must be increased if the efficiency is to be improved, and that can be done not only by increasing the pipe diameter, but also by improving the scheme of switching over the valves.

The improved scheme (Fig. 7) aims at making full and immediate use of the state of pressure and velocity created by the passage of a wave; and so it is arranged that a valve operates as soon as the wave has travelled once along the pipe and not, as in Scheme I, when it has completed the return journey. In one cycle of operations in the new scheme, the wave travels four pipe lengths compared with six in the other. The period of the cycle is reduced to 2/3 of its former value, while the delivery per cycle remains the same. At the beginning of the first phase the pressure in the pipe is, from (3),

$$p_{min} = p_M - p_H + p_N \quad .. \quad .. \quad (8)$$

and is, therefore, 1 atmos. in the example given in Fig. 7.

The four phases in the cycle of this example are made clear by the following table and Fig. 8.

Phase	At start of phase: Pipe pressure	At start of phase: Direction of flow	Pipe ends: A connected to	Pipe ends: B connected to	Direction of flow in pipe: at A	Direction of flow in pipe: at B	Direction of wave
1	1 atmos.	—	M.D.	—	In	—	Right
2	5 ,,	Right	M.D.	H.D.	In	Out	Left
3	7 ,,	Right	N.D.	H.D.	Out	Out	Right
4	3 ,,	Left	N.D.	—	Out	—	Left

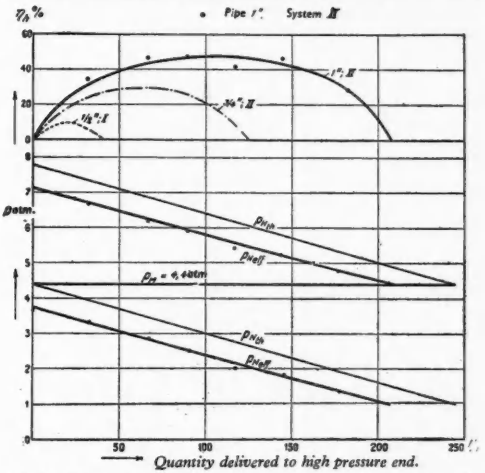


Fig. 9. Lower part: Pressures in theory and as measured; Upper part: Hydraulic efficiency using various sizes of pipe. Schemes I and II (l/h = litres per hour, $\eta h\% =$ Hydraulic efficiency in % P_{Nth}, P_{Neff}, P_{Hth}, and P_{Heff} = Theoretical and actual pressures).

The quantity taken from the I.P. tank, M.D., in Scheme I is $\Delta V = 1F\ (p_{max} - p_{min}) / \rho a^2$ *per cycle.*

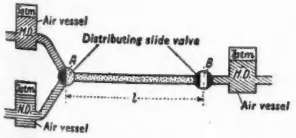


Fig. 7. Scheme II (Items as for Fig. 3).

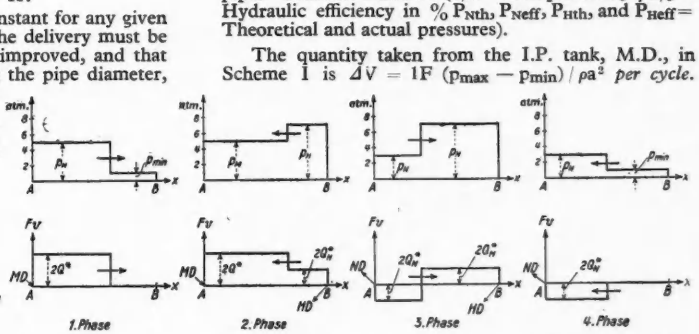


Fig. 8. Diagram of waves of pressure and velocity (Scheme II).

In Scheme II, the quantity taken in the first phase is $1F(p_M - p_{min})/\rho a^2$; and, in the second phase, since there is no change in the velocity of influx at A (see Fig. 8), the quantity is the same as in the first phase. Therefore, by (8) and (3), the quantity *per cycle*, is

$$2\frac{1F}{\rho a^2}(p_H - p_N) = \frac{1F}{\rho a^2}(p_{max} - p_{min})$$

which is exactly the same as in Scheme I.

The frequency is now $n^* = \frac{a}{41}$ cycles per second.

The quantity delivered *per second* is, therefore,

$$Q^* = n^* \Delta V = k^*(p_M - p_{min}) = k^*(p_H - p_N)\ m^3/sec.$$

$$\text{where} \quad k^* = \frac{F}{2\rho a} \quad .. \quad .. \quad .. \quad (9)$$

In the fourth phase the volume flowing into the L.P. vessel, N.D., is $1F(p_N - p_{min})/\rho a^2$, which is the same as in the third phase. The rates of flow to the L.P. and H.P. vessels are, therefore,

$$Q^*_N = k^*(p_N - p_{min}) = k^* \Delta p_H\ m^3/sec.$$

$$\text{and } Q^*_H = Q^* - Q^*_N = k^* \Delta p_N\ m^3/sec.$$

$$\text{where } \Delta p_H = p_H - p_M, \text{ and } \Delta p_N = p_M - p_N \quad .. \quad (10)$$

Test Results.

A test model built to the design of Prof. Eichelberg at the Federal Technical College gave valuable information for the design of full-scale developments.

The valves A and B, in a common housing, were operated by a belt-driven D.C. motor. The air vessels were directly connected to the valve housing to make the connecting pipes as short as possible; but, even so, the pipe friction was enough to cause some flattening of the wave front. Manometers were fitted to the air columns of the air vessels, and indicators were connected to the ends of the experimental pipe. This pipe was in the form of a bow and measured 14.47 ms. between valve centres, the time taken by a wave to travel along the pipe being about 1/100 sec.

The I.P. air vessel was connected to the water main of 4.5 atmos. pressure ($= p_M$); and, in the pipe, p_{min} was made 1 atmos. These pressures give the following:—

$p_M - p_{min} = 3.5$ atmos.
$2p_M - p_{min} = p_{max} = 8$ atmos.
$\Delta p = p_{max} - p_{min} = 7$ atmos.

Equation (1) makes $Q = 0.0785 \times 3600F = 282.2F$ m.³/hour for Scheme I. The rate of delivery in Scheme II is 50% more; so $Q^* = 1.5Q$. Tests were made with pipes of ½", ¾" and 1" diameters, for which the following are the theoretical values:—

Pipe dia. inches	Area F 10^{-4} m²	Q lit./hr	Q* lit./hr
½	1.265	35.8	53.7
¾	2.85	80.7	121
1	5.07	143.6	215.4

The measured quantity, using a 1" dia. pipe and Scheme II, was $Q^* = 245$ lit./hr. ($Q^*_H = 105$ lit./hr. and $Q^*_N = 140$ lit./hr.), or 14% more than was calculated. The explanation is that water is fed in by the rotation of the valves.

Before starting up, the H.P. vessel was put under a certain initial pressure p_H by means of compressed air, and the appropriate pressure p_N at the L.P. end was obtained by means of an outlet valve. As soon as the control valves reached their correct speed the pressures built up automatically. To increase output the outlet from the H.P. vessel was altered with consequent change in the values of p_H and ν. The pressure in the L.P. end adjusted itself to the appropriate value.

Equation (7) shows that Q_H and Q_N must not be too small if the desired efficiency is to be reached. The tests gave best efficiency η_h when approximately half the theoretical Q was delivered to the H.P. end. The improvement in η_h by increasing Q, whether by enlarging the pipe or by changing over from Scheme I to Scheme II, is demonstrated in Fig. 9, which also shows the pressures when the 1" dia. pipe was used.

The maximum efficiency of 48% was measured with a Q_H of 105 lit./hr., or 2.92×10^{-5} m³/sec., and a Q_N of 140 lit./hr., or 3.89×10^{-5} m³/sec.; and p_H was 1.32 atmos., or 13,200 kg./m², and p_N 2.08 atmos., or 20,800 kg./m². The power taken by the valves was about 0.8 kW., i.e., about 80 kg.metres per sec. The overall efficiency was, therefore,

$$\eta_{tot} = \frac{0.292 \times 1.32}{0.389 \times 2.08 + 80} = 0.47 \textit{ of } 1 \textit{ per cent.}$$

This disappointing result is, however, due to the unfavourably low output in comparison with the mechanical power required for the valves of the small test apparatus. This means that an improvement of the ram is practicable only if the capacity is very substantially increased. Pipe areas must be large compared with the valve passages in order to minimise the driving power required and the losses due to leakage. But, above all, the output can be raised by increasing the pressure ranges; for Q increases with Δp, whereas leakages are proportional to $\sqrt{\Delta p}$.

The chief trouble encountered in the tests was to keep the valves tight, particularly in the direction of rotation, because, in the absence of intermediate packings, some water is driven round and passes from one valve passage to the other.

Application to Storage Schemes.

The model tests clearly show that the ram is economically feasible only in the largest sizes where the water quantities are comparable with those handled by storage pumps. For example, in a water power scheme where the capacity to store the available flow is inadequate and water in excess of demand is delivered to an accumulation basin at a higher level, by means of pumps, the same duty could be performed by the new invention with the advantage that it is direct and without the complications entailed by the use of electrical energy.

Fig. 10 shows the invention applied to a scheme of that kind. At times of large power demand, No. 1 turbine can run at full capacity under a pressure of 50 atmos., and the output to the line can be augmented by No. 2 turbine fed from the accumulation basin. If $Q = 1$ m³/sec., the power available to No. 1 turbine is

$$\frac{\gamma QH}{75} = \frac{1000 \times 1 \times 500}{75} = 6670 \text{ metric H.P.}$$

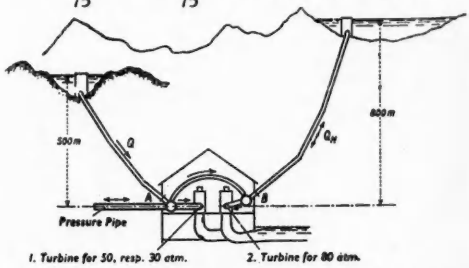


Fig. 10. Hydro-electric scheme with the Eichelberg ram used as a storage pump.

During periods of low demand, the quantity Q would be dealt with by the ram and separated by it into Q_H at a pressure of 80 atmos., and Q_N at 30 atmos. Q_N could be used to generate power from No. 1 machine while Q_H would go to the accumulation basin.

Neglecting losses $\quad \dfrac{Q_H}{Q_N} = \dfrac{\Delta p_N}{\Delta p_H} = \dfrac{20}{30}$

Therefore .. $Q_H = 0.40$ m³/sec. and $Q_N = 0.60$ m³/sec.

Accordingly, the ram gives the following :—

Output to accumulation basin $= \dfrac{400 \times 800}{75} = 4270$ metric H.P.

Output to No. 1 turbine $= \dfrac{600 \times 300}{75} = 2400$ metric H.P.

Total .. 6670 metric H.P.

For this example Prof. Eichelberg has calculated the efficiencies η_h and η_{tot} to be approximately 0.8 and 0.74 respectively on the basis of the measured losses due to leakage and friction in the tests of the model. On the other hand, if electrically-driven pumps are used, the efficiency of pumping from A to B, Fig. 10, is

η tot $= \eta$ turbine $\times \eta$ generator $\times \eta$ motor $\times \eta$ pump,

which is only about 0.66.

(*To be continued*).

THE CONTROL OF AUTOMATIC CONVEYING PLANTS

By G. ROCHAT. (From *Brown-Boveri Review*, Vol. 29, No. 5, May, 1942, pp. 129-137.)

FOR the automatic control of conveying plants there are two different fundamental principles entailed. The first meets the conditions necessary for automaticity by a suitable combination of various control devices which act directly on the apparatus controlling the motor. This solution implies a choice of apparatus with a large number of contacts, in order to meet the needs of the numerous interlocks and connections between the different parts of apparatus and a complicated diagram of connections.

The second principle has been developed for all combined installations, which are those in which there are several motors which have to run alternatively, or the speeds of which have to be varied in function of a programme laid down in advance. This is the programme-switch principle. A central control apparatus which gets the requisite impulses from the manœuvring switches, in its turn imparts to the contactors and relays the successive impulses corresponding to the desired programme. The programme switch (Fig. 1) ensures the proper succession of the various operations, and thus is the "brain" of the automatic control. Not only does it allow of equipping the whole plant with simple manœuvring switches, but it permits also of trying out and tuning the complete plant in the shops. This allows of reducing tests on site to a minimum.

The programme switch is designed with a servo-motor which drives a shaft with cams mounted upon, each of which governs a contact of ample dimensions. The servo-motor is of the vibrating-armature type, having neither commutator, brushes nor flexible leads. In most cases, the programme switch controls the plant through the intermediary of contactors.

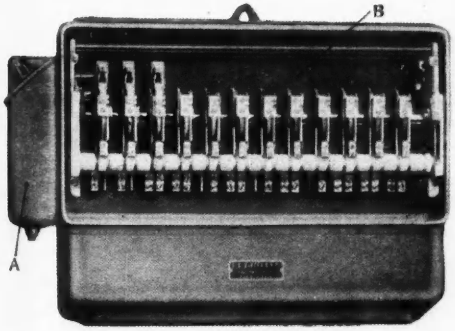


Fig. 1. Programme switch (cover removed). A, Oil-tank of auxiliary motor; B, Hammer type contacts actuated by cams.

A characteristic automatic conveying plant is the coke elevator in the Geneva municipal gasworks (Fig. 2). Hinged-bottom buckets mounted on motor trucks are loaded with incandescent coke supplied from the ovens, and travel to the foot of an extinction tower. Here they are taken over by an inclined elevator track, and are carried up to a position above the loading aperture in the upper part of the tower; they are then emptied into the extinction chamber. The empty bucket now descends to the motor truck, which is then ready to fetch another load.

The mechanical part of the equipment comprises a metal structure resting on the concrete tower, and on which runs a crab with hoisting gear. The loaded buckets coming to the foot of the tower are suspended to a beam which has wheels engaging in the rails of the inclined elevator.

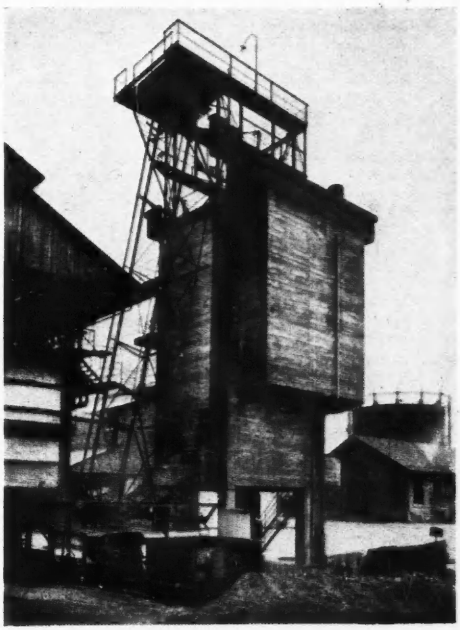
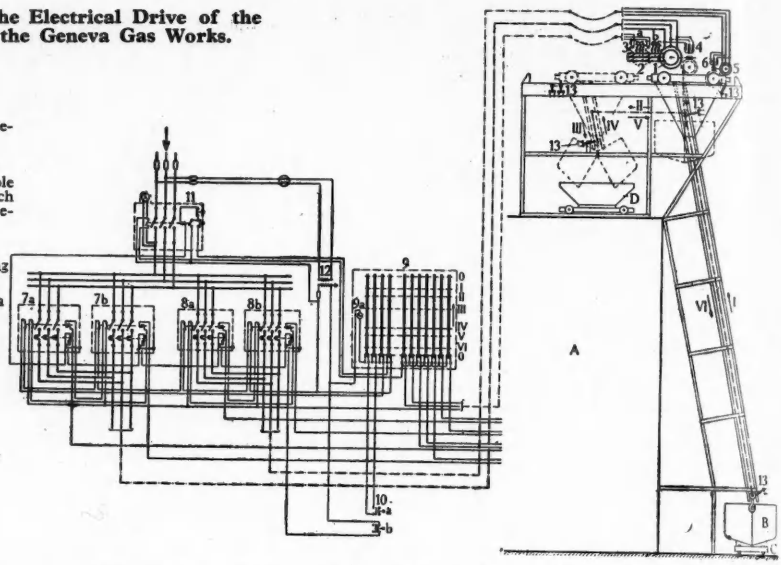


Fig. 2. Dry-coke extinction plant of the Geneva Gas Works.

Fig. 3 **Diagram of the Electrical Drive of the Coke Elevator in the Geneva Gas Works.**

1. Winch motor
2. Starting resistances
3a, b. Rotor contactors
4, 6. Three-phase brake-releasing magnets
5. Crab motor
7a, b and 8a, b. Three-pole reversing contactors of winch motor and crab motor, respectively.
9. Programme switch
9a. Programme switch driving motor
10. Push button switch; a "in," b "out"
11. Main switch box
12. Single-phase auxiliary transformer
13. End-travel switch
A. Cooling tower
B. Coke bucket
C. Bucket-conveying motor truck
D. Filling hopper



The plant is supplied with three-phase current 380 V., 50 cycles. The electrical equipment comprises a 26 kW. slip-ring type hoisting motor and a 1.5 kW. squirrel-cage travelling motor for the crab. Fig. 3 shows in simplified form the connections.

The winch motor 1 is switched in for hoisting or lowering by contactors 7a or 7b. It starts with the help of resistance 2, the two steps of which are short-circuited one after the other by contactors 3a 3b. The motor 5 displacing the crab is controlled by contactors 8a and 8b. The end-travel positions of bucket and crab are limited by the end-travel switches 13. Main switch 11 allows of cutting out the whole plant. The various contactors are controlled by the programme switch 9.

Let us assume that bucket B has just been secured to the carrying beam. By depressing push-button 10a, the operator switches in the programme switch, the drum of which revolves from position O to position I. During this rotation hoisting contactor 7a and the contactor 3a and 3b close one after another. The bucket moves upwards. When it reaches its highest point it causes the corresponding end-travel switch to act, which cuts out contactor 7a. The programme switch, having got a new current impulse from an auxiliary contact of contactor 7a, rotates from position I to position II. In this position contactor 8a closes and the crab motor is put under voltage, and the crab with the bucket moves towards the filling hopper D. The lowering and raising of the bucket, its return journey and the descent, are effected by a similar process. At each stop brought about by the corresponding end-travel switch, the programme switch 9 gets a new current impulse and moves to the next position.

Control of the whole plant is very simple, as a single depression of push-button 10 suffices to bring about a complete circle comprising six different movements.

A NEW HIGH-PRECISION METHOD FOR SHORT-CIRCUIT MEASUREMENTS ON TRANSFORMERS

By Dr. P. Waldvogel. (From *Brown-Boveri Review*, Vol. 29, No. 5, May, 1942, pp. 126-129.)

The transformer is distinguished by the great precision with which its characteristics can be calculated in advance. One of the results of this is that testing engineers are constantly obliged to improve methods of measurement, because the margin allowed by the tolerance between calculated and measured magnitudes is so narrow.

This applies especially to short-circuit tests, the object of which is, on the one hand, to determine the impedance voltage and, on the other, to determine the copper losses under load (i.e., the pure resistance losses and the stray eddy-current losses).

In all standard transformer designs, the magnetic field which forms under short-circuit conditions exists chiefly in the spaces between the high-tension and low-tension windings, because the windings are composed of ampere-turns in opposition. Obviously these magnetic lines of force close outside the windings, either through the air, or through the laminations, but the flux density in these regions is very low. For this reason the reluctance of the field is really confined to the air; in other words, the reluctance and, therefore, the inductive component of the short-circuit impedance are constant values which remain independent of the strength of the current flowing in the windings during the test.

The resistance component of the impedance voltage depends on the pure resistances and the additional resistances of the eddy-current circuits. The pure resistances are quite independent of the current. This is also true of the additional resistances, which owe their existence to a magnetic field uninfluenced by the degree of saturation.

Consequently, there is a strictly linear relation between voltage and current during a short-circuit test in any standard transformer. It is logical to choose a

measurement current which is most suitable from the point of view of measurement requirements alone instead of the normal full-load current.

Obviously a bridge connection seems the best solution for carrying out measurements. The bridge is supplied with a voltage the magnitude of which is determined solely by the zero-indicating instrument. There is, however, identity of measurement frequency and rated transformer frequency to be strictly observed. An absolutely perfect supply voltage curve is not indispensable, if the zero-indicating instrument has a sufficiently sharp resonance at the given frequency. For example, a vibration galvanometer is quite suitable.

As a rule, the modified Wheatstone bridge connection for alternating current measurements, shown in Fig. 1, is used. For example, resistance r is regulated and capacity c (that is, the capacitive reactance $x_c = \frac{1}{\omega C}$) until the galvanometer G indicates the exact zero. Then

$$\frac{R+jX}{q} = \frac{p}{r-jx_c}$$

is satisfied, where R+jX (complex notation) represents the measured value of the impedance of the short-circuited transformer. When the real and imaginary parts are separated, this leads to

$$R = \frac{pq}{r^2 + X_c^2}\, r \qquad X = \frac{pq}{r^2 + X_c^2}\, x_c$$

As soon as R and X are known, it is an easy matter to find the active and reactive load corresponding to the operation of the transformer in short circuit.

The Wheatstone bridge cannot be recommended when the impedance R+jX is very small. In this case the unavoidable contact resistances cannot longer be ignored. The modified Thomson double bridge can be used to advantage (Fig. 2). As is known, the influence of the contact points is eliminated with this bridge connection.

We have used as zero-indicating instrument both a vibration galvanometer and a cathode-ray valve apparatus, built by Messrs. Trub, Tauber & Co. of Zurich, containing a valve similar to those used in radio receiving sets as resonance indicators.

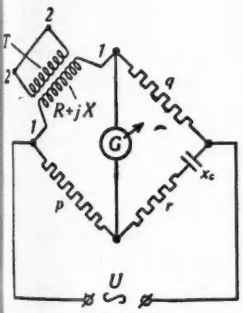


Fig. 1. Modified connection diagram of a Wheatstone bridge for measuring the impedance of a short-circuited transformer

T Short-circuited transformer
1-1 Primary terminals of T
2-2 Secondary terminals of T
R+jX Short-circuit impedance of T, referred to primary side
p, q, r Resistances.
$x_c = \frac{1}{\omega C}$ Capacitive reactance.
G Zero instrument.
U A.C. supply voltage

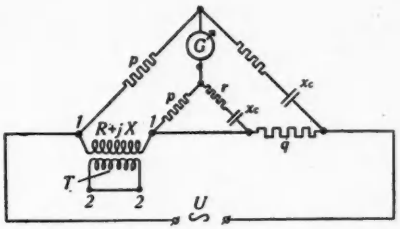


Fig. 2. Thomson double bridge in modified form for measuring the impedance of a short-circuited transformer, when the impedance is low. (Items as for *Fig.* 1).

As the bridge can only be used in single-phase connection, the difficulty in measuring three-phase transformers can be overcome by carrying out three successive measurements on terminals UV, VW and WU. This process is fully justified by the principle of superposition.

The transformer testing department of Brown-Boveri & Co. is equipped with an impedance bridge for carrying out short-circuit measurements. It is supplied through a transformer of 40 VA rating only, and allows of carrying out tests with the greatest facility measurements of a precision unattained up till now on transformers up to 40 MVA.

On the basis of statistics drawn up from the results of about thirty measurements, it was proved that there was no fundamental difference between the results obtained by the usual method with voltmeter, ammeter and wattmeter and those with the new method, both as regards impedance voltage and short-circuit losses. The characteristic feature of the new method is that measurement repeated on the same transformer or on several identical transformers show considerably more constant results than those carried out with the usual method. This is explained, on the one hand, by the higher sensitivity and, on the other, by the elimination of the influence exercised by temperature.

The bridge method is exceptionally suitable for carrying out short-circuit measurements on transformers of big outputs and very high voltages, because these units have a very low power factor in short circuit (for example, 0.05 and less). Under such conditions, there would be big differences in using a wattmeter. For some time, the measurement of the voltage ratio has been carried out in potentiometer connection with voltage supply from the low-voltage side (220 V.). The bridge described here completes this apparatus in a most desirable manner, because in future it will be possible to measure with great accuracy the voltage ratio, impedance voltage and the copper losses of the biggest transformers. This also applies to all the magnitudes influencing the possibility of parallel operation. Further, all these measurements can be carried out with the aid of a low-voltage source of current with an output of only a few volt-amperes. Finally, only the no-load test demands an appreciable source of power.

CONTRIBUTION TO THE METALLURGICAL TREATMENT OF MALLEABLE CAST IRON

By Guenter Brinkman and Paul Tobias, Research Laboratory Heinrich Lanz A.G. Mannheim. (From *Die Giesserei*, Vol. 29, No. 21, October, 1942, pp. 356-358.)

For the production of high-quality malleable castings it is essential that the period of annealing should be preceded by an appropriate melting technique.

It is known that in the process of malleable iron manufacture the cementite is converted into temper carbon during the period of high-temperature treatment. This annealing time can be reduced for any given temperature by the addition of elements which encourage graphitization, e.g., silicon or copper. G. Brinkman and P. Tobias, however, have shown that the shortening

can alternatively be effected by refinement of the grain, without the necessity of changing the chemical composition.

Both in black-heart practice and in the production of white malleable cast iron decarbonization also occurs during the period of heating. It is the velocity of this decarbonization which determines the length of the annealing time required.

The influence of the chemical composition of the pig iron on the duration of the time of annealing has not yet been determined.

It is important in the tempering process to avoid premature decomposition of cementite, as the free carbon so produced is more difficult to gasify. On the other hand, a residue of cementite on completion of decarbonization unnecessarily prolongs the annealing period and so is economically undesirable. From this it is clear that the breaking down of cementite must be carefully correlated with the decomposition time.

For black-heart work, with the same chemical composition as black cast, a large grain is desirable. White malleable cast iron, however, demands a coarser grain and a relatively small silica content corresponding to its prolonged decarbonization period.

In white malleable cast-iron work it is also important to ensure that the nuclei are sufficient in number. If these are too few, the crude iron will be brittle and sensitive to impact, and the castings may tend to break in the moulds. The condition also favours cementite residues, thus prolonging the annealing time.

P. Bardenheuer has suggested that the nuclei consists of SiO_2 silicates of iron and sulphides in very fine dispersion. The nuclei can be removed from the melt by the use of slags which are zero in metallic oxides or which have reducing properties or alternatively by extension of the cooling process over a long period.

C. H. Morken has demonstrated that iron melted in a reducing atmosphere is free from nuclei. This iron may have a high silica content without danger of primary graphite elimination. Honda and Murakami claim that iron melted in a vacuum is extremely difficult to graphitize, owing to its freedom from nuclei.

The actual number of nuclei present depends on the charge and on the type of furnace used. Grain size is influenced by the velocity of decomposition or by the addition of elements such as aluminium or titanium. Refinement of grain is effected by the use of deoxydizing agents such as aluminium.

The production of high-quality malleable castings in various types of furnaces may now be considered.

The Cupola Furnace.

Charges melted in the cupola furnace are rich in nuclei, owing to the high temperature involved and to its contact of the iron with the furnace gases. Those melts are best suited for black-cast and black-heart work. The main disadvantages of the cupola furnace are the high carbon (2.8%) and high silica content of the iron, and the inadequate control of the composition.

With a reasonably low silica content, cupola furnace iron, in spite of its numerous nuclei, can also be used for white malleable casting. A prolonged cooling in the fore-hearth will considerably reduce the number of nuclei.

The Reverberating Furnace.

This produces a practically nuclei-free iron in consequence of its long period of melting at a comparatively high temperature, and the large area of contact between slag and bath. The choice of the charge is of supreme importance, especially in the Siemens-Martin furnace. An unsuitable mixture may produce an iron which develops cavities and tends to burst. The Siemens-Martin furnace is unsuitable for the production of black iron and black-heart work.

The Bracklesburg Furnace.

This is ideal for the production of black cast, as the iron is extremely rich in nuclei, particularly fine ones being produced by the final rotation of the furnace. With the help of a little oxidation the melting is very quick. The bath absorbs very little ferrous iron and therefore does not come to the boil. This type of furnace is especially suitable for thin-walled black casting, owing to the low available carbon content of the melt, and to the super heating.

The Electric Arc Furnace.

It has not yet been found possible to produce a satisfactory malleable casting in this furnace. The lack of success is probably due to removal of gas by the excess heat. Poor tempering probably results from the absence of nuclei in the melt, and can be remedied by the addition of aluminium. This metal, used in quantities up to 0.06%, also increases elongation.

Diagram 1 shows the effect of the addition of 0.1% aluminium on the nuclei content of two melts, determined by the formation of temper carbon after an annealing period of six hours at 950°, followed by a graduated cooling time. In the nuclei-free melt the number of nuclei and consequently the temper carbon formation increases with the standing time after the addition of aluminium. In the melt containing nuclei the addition of aluminium causes a preliminary decrease in their number with lengthening of the standing time, followed by a subsequent increase. With further prolongation of standing time the nuclei content is diminished by agglomeration. To achieve a given nuclear concentration it is advisable to start with a nuclei-free melt such as an electric melt, or to deoxydise completely with silicon.

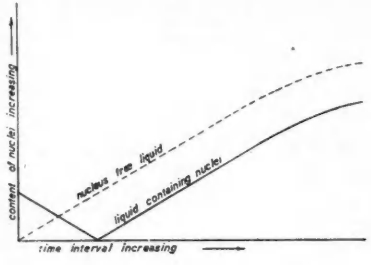


Fig. 1

Summary.

The required period of high-temperature annealing is proportional to the velocity of decomposition of cementite, which depends on chemical composition and on grain size. In black-heart or white malleable casting, the rate of decarbonization also influences the length of the annealing time. Early decomposition of cementite is undesirable, owing to the difficulty of redissolving temper carbon. Graphitization time must, therefore, be adjusted to decarbonization, either by appropriate chemical composition or by varying grain size. Nuclear content, and therefore velocity of decomposition, is dependent on the charge, the type of furnace, and the process used. The nuclear number can be precisely controlled in the bath, particularly if the initial melt is free from nuclei.

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- NEW EQUIPMENT
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- NOTES

NEW EQUIPMENT

IMPROVED METHODS OF OBTAINING SLOW SPEEDS ON ELECTRIC OVERHEAD TRAVELLING CRANES ON A.C. SUPPLIES.

On account of the general extension of alternating current supply which has taken place in this country during the past few years, a demand has arisen for methods of controlling the hoisting motion of overhead electric cranes on A.C. Further, in view of the large number of women now employed in factories and the increased speed of production "controllability" of overhead cranes assumes an enhanced importance. When lifting heavy pieces of machinery in and out of machine tools or assembling them together, it is necessary to be able to move the load slowly and gently without jerking, otherwise accidents occur. Output can be speeded up considerably if crane drivers and loaders know that the loads they are handling can be moved slowly and gently, or be run up to full speed without loss of time, if desired. Until recently it was believed that only a D.C. supply could give the following characteristics :—

(a) Lifting or lowering light or heavy loads at a steady or creeping speed, with a choice of several slow speeds.

(b) Starting gently from rest and accelerating rapidly up to full speed.

(c) Dynamically braking the descending load and preventing it attaining a dangerous speed without mechanical brakes.

(d) "Off" position dynamic braking preventing the load from descending at more than creeping speed in the event of total failure of the mechanical brakes.

(e) Lighter loads handled at higher speeds by virtue of series characteristics of the motor.

These features are demanded by cranes serving foundries, heavy machine shops, assembly bays, power stations, etc., and numerous schemes using alternating current supplies have been applied to meet these conditions with varying degrees of success. It is proposed to describe two of the latest schemes now available for meeting this increasing demand.

Advent of the long speed range A.C. commutator motor, which in conjunction with a suitable controller has been successfully applied to crane work, fully satisfies characteristics (a), (b) and (c); it does not fulfil (d) and (e) but has introduced compensating advantages not found on any D.C. equipment.

The motor used is a 3-phase, variable speed A.C. commutator motor either screen protected or totally enclosed, fitted with pilot motor operated brush gear. The speed range is usually arranged between 10:1 and 25:1 according to circumstances. Any speed within the range can be readily obtained, the equivalent of a controller with an infinite number of notches. The controller itself is of the airbreak drum reversing type, operated by crank handle and of the standard type generally used on cranes.

The controller has three notches in each direction and on the first notch the motor is connected across the line with the brush gear at the slow speed position giving slow running. The third notch gives full speed by allowing the pilot motor to move the brush gear round to top speed position. The second notch of the controller leaves the main motor connected to the line, but disconnects the pilot motor and so enables any intermediate speed between slow and fast to be obtained. Thus, if the controller is moved on to notch three, and as soon as the desired speed has been obtained on the motor, moved back to notch two, this speed will be maintained indefinitely. In the same way, if the motor is running at full speed, the controller is moved back to notch one, the motor will slow down and any intermediate slow speed may be maintained by again moving the controller to notch two. This gives a very flexible form of control and one which is very easily understood after a few minutes' practice.

Electrical interlock gear renders the system foolproof and prevents the motor being reversed before the brushgear has moved to the slow speed position.

The following features are peculiar to this system :—

(A) No controlling resistance is needed, thus eliminating resistances losses.

(B) Shunt characteristic of motor gives safe control of descending loads at the speed required by the operator, and load cannot race away or get out of hand.

(C) The speed of handling light or heavy loads varies only slightly on a particular brush setting.

(D) Practically the whole of the power taken from the line is productive and is proportional to the motor output. In the absence of resistances, power savings are greatest at the low speed which may be maintained for any length of time without overheating the motor.

(E) Starting torque available on the motor is more than sufficient to lift the heaviest load which the crane, itself, is capable of handling with safety.

(F) Since the acceleration of the motor is governed by the speed at which the pilot motor moves the brush gear from slow to fast, the controller may be moved directly from "off" to notch three in either direction, and the motor will lift or lower its load on the hook smoothly and safely in the shortest safe time. In effect the controller may be used as a plain reversing switch without harm to the motor and ensuring speedy operation of the crane.

(G) A considerable saving in wear of the brake linings of the standard electro mechanical brake, with which the crane is fitted, can be achieved by employing the dynamic braking feature of the motor for bringing the load to rest from full speed. This, of course, is achieved by bringing the controller back on to notch one, causing the motor to decelerate to a creeping speed and then switching off, when the electro mechanical brake operates and brings the load finally to rest. Prolonged tests on cranes in service have proved that the additional heating caused by the dissipation of energy in the motor when decelerating, rather than in the brake shoes, has no deleterious effect on the motor itself, and does not cause overheating.

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(H) The motor is designed to be completely non-self exciting at any brush setting or under any conditions of brush movement. This important point is necessary when it is considered that the electric brake solenoid is connected in parallel with the line terminals of the motor, and must, therefore, apply the brake immediately the line current is cut off. Any tendency towards self excitation would, of course, cause the brake solenoid to stick up, with resulting damage.

In the hoisting direction rotor resistance control of a standard slip ring motor, enables the load to be lifted at slow or creeping speeds. In the lowering direction the torque produced by the weight of the descending load due to gravity is balanced in varying degrees by an opposing torque provided by the hoisting motor, giving lowering speeds from a creeping speed to normal full speed. An overspeed device is provided to prevent the descending load from exceeding a predetermined safe value under all circumstances, thus rendering the control gear mistake-proof, even in the hands of an unskilled operator. The following components are used in this scheme.

The controller is of the airbreak drum type, having 12 notches and provided with a crank handle incorporating a spring loaded switch, thus enabling instant application of the brake to the load at any position of the controller. For floor controlled cranes, the spring switch is incorporated in a special hand grip type push button, suspended at one end of the controller chains at floor level. There is thus no " off " position on the controller.

A panel containing two triple pole reversing contractors is provided for controlling the stator of the hoisting motor, together with a time delay relay operated by an overspeed device.

Resistances are of the standard 3-phase unbreakable grid or coil type, liberally rated and suitable for slow speed up or down.

The operation of the controller is very similar to that of the standard drum type controller found on most travelling cranes, but with one important exception. A spring switch is incorporated in the operating crank handle controlling the main contactors, so that the load may be instantly brought to rest on any notch of the controller without moving the handle, but simply by the driver releasing the pressure of his hand on it. Moreover, the load can be restarted in a lowering direction by simple pressing the crank handle, the load continuing to move at the speed at which it was travelling before being brought to rest. Provision is made, however, to prevent the driver starting in the hoisting direction without sufficient resistance being in the rotor circuit of the motor, to prevent the flow of heavy currents which may damage the rotor.

The notches on which the load can creep up or down are determined by the amount of load on the crane hook. Where the driver needs to make delicate movements of the load, usually he can work on three notches, the loads being held without movement on the centre of these three notches, without releasing pressure on the crank handle. Thus the effort to drive the crane is very considerably reduced as there is no need for the driver to move the controller back to an " off " position each time he wishes to stop the movement of the load.

This results in less wear and tear on the controller, and on the brake mechanism on the hoist motion, since the load can so easily be brought to rest by movement of the controller without having to depend on the closing of the brake sheaves.

On the " lower " notch of the controller, a very light load or the light hook may be lowered swiftly, simply by depressing the controller handle, which causes the hoisting motor to drive down. Should a very heavy load be inadvertently lowered in this way, it would be instantly brought to rest after a few feet by the overspeed device, before there was any possibility of damage being done.

Referring back to the list of D.C. characteristics, given at the beginning of this article, it will be noted that this second scheme fulfils conditions (a) and (b); (c) is fulfilled in that the load cannot get out of control while lowering, and yet can be lowered at the highest safe speed to suit the mechanical design of the crane, which is a feature not obtainable on the commutator motor scheme.

This assists in maintaining speed of production and is comparable to (e); the absence of this feature is a common cause of disappointment to users of cranes on A.C., especially if their cranes have been converted from D.C. to A.C., in a general change-over of supply.

It is now apparent that many of the arguments for cranes on D.C. supplies are brushed aside by the introduction of these two systems and as the general extension of A.C. has become a national policy it is not in the country's interest to retain D.C. equipment mainly to operate cranes as is quite often the case when a change-over is carried out. It is very essential at the present time that a manufacturer's existing standard equipment should be used wherever possible, to avoid expense and special new gear being developed and manufactured; and it is, therefore, worth noting that the two control systems described in this article embody manufacturer's standard products throughout, and no special equipment has to be constructed for the purpose of obtaining the results which have been described.

The systems described above are covered by Vaughan Crane Company's Patents 409485, 528926 and 540346.

PERSONAL

Mr. E. J. Pode has been appointed Assistant Managing Director of Guest Keen Baldwins Iron and Steel Co. Ltd.

Mr. Hans Renold, founder of Hans Renold Ltd., a company now embodied in the Renold and Coventry Chain Co. Ltd., has died recently.

BUSINESS CHANGES

Wearsten & Guylee Ltd., have changed the name of the firm to Frank Guylee & Son, Ltd., Archer Tool Works, Millhouses, Sheffield, 8.

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